THE

MARINE STEAM E

ITS

CONSTRUCTION, ACTION, AND MANAGEMENT.

A MANUAL AND BOOK OF REFERENCE

FOR

ENGINEERS, OFFICERS OF THE NAVY AND MERCANTILE MARINE, PRACTICAL MECHANICS, STUDENTS OF TECHNICAL SCHOOLS, SHIPOWNERS, AND ALL INTERESTED IN STEAM NAVIGATION.

WITH AN ATLAS OF LITHOGRAPHIC PLATES
CONTAINING 1500 FIGURES EXECUTED FROM WORKING DRAWINGS COLOURED.

BY

CARL BUSLEY.

THIRD EDITION, THOROUGHLY REVISED AND ENLARGED.

TRANSLATED BY H. A. B. COLE, M. I. N. A.

VOLUME I.

TEXT.

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Preface to the Third Edition.

In 1891 when the first portion of Vol. I. appeared it was certainly not my intention to defer the completion of it till now. Had I then foreseen how little free time would be at my disposal for transforming the earlier editions the third would never have been undertaken at all. In my present position, which no longer affords me the leisure for comprehensive study I formerly enjoyed as a teacher, and hinders the acquisition of suitable matter for the atlas of plates, I can, to my genuine regret make no binding promise to the possessors of the portions already published as to the date of completion of the entire work. All engineers engaged in practice will understand and appreciate the difficulties perpetually confronting me, more especially in the preparation of the plates and in keeping them above the level of a mere *rechauffé* of drawings already published in the technical periodicals. I hope however that the treatment of the theory of the steam engine and the construction of marine boilers will be found sufficiently exhaustive to obtain popularity for the book as it stands, the more so as a general index and an index of the names referred to are appended to the first volume.

The very brief extract from the mechanical theory of heat in the first edition has now been so extended as to render the third division which is new, intelligible. This division is devoted to modern investigation of the steam engine, giving due prominence to the influence of the cylinder walls on the steam, a more thorough knowledge of which has chiefly shaped the path of advancement in marine engineering. An attempt has been made to compile as tersely as possible the records of various researches from the technical journals of the countries interested

in engineering. Investigations based upon the indicator, which *faute de mieux* still furnishes the most useful data, have received thorough treatment.

The fourth division devoted to fuels, has been entirely remodelled. The process of combustion and the losses of heat in the still far from perfect furnaces of marine boilers, as well as the extreme importance of the capability of the firemen have been throughly gone into under the light of recent experiments. Liquid fuels have been comprehensively treated of on account of their extending application to warships.

The fifth division is practically identical with the former fourth division. It relates to the performance and economy of marine engines but upon a much more extended basis than before as befits the development of the multiple-expansion principle.

The sixth division is partially compounded of the former sixth and twenty-second divisions. It has been thorougly-revised and in its present form will be found practically useful in estimating the I. H. P. for a proposed steamer especially by means of interpolation in the table of actual recent performance-coefficients. The selection of a suitable type of engine will be facilitated by the seventh division which contains numerous results of compound and multiple-expansion engines.

The eighth division has been enlarged where it treats of triples and quadruples. It is intended to show the inexperienced how simply by means of Mariotte's law, a diagram can be constructed to represent the variations of pressure in the cylinders and of the turning moments as well as to enable the degree of uniformity of the working of the engine to be pre-judged. The applicability of Mariotte's law is specially referred to on page 131.

The various types of marine boilers are very much more exhaustively discussed in the ninth division than in the former fifth division. The most notable water-tube boilers are closely described and it is pointed out that as yet no final judgement can be arrived at as to the superior fitness of any one system for marine purposes.

The rules of the principal Classification Societies which relate to the construction of marine boilers have been brought together and the measurements all reduced to the metrical system thus simplifying calculation and facilitating rapid comparison.

In the eleventh division under the heading of furnaces the various arrangements of induced and forced draught have received due attention.

My best thanks are due to Herr Marine Oberbaurath von Jaski for his excellent treatment of the final two divisions of the volume, tke twelfth & thirteenth, on boiler mountings and the disposition of boilers in the ship. He has exhaustively described modern boiler mountings adapted to high pressures and discussed the difficult question of boiler feeding.

The publishers have striven to maintain the same high standard as before in the get-up of the book and the atlas of plates.

Berlin, November 1900.

The Author.

Contents of Volume I.

First Part.

General Theory of Marine Engines.

First Division.

Principles of the Mechanical Theory of Heat.

Second Division.

Steam.

Third Division.

The process in the cylinder.

Fourth Division.

Fuels.

Fifth Division.

Determination of the Power & Economy of Marine Engines.

Sixth Division.

Estimation of the power required for proposed steamers.

Seventh Division.

Selection of the most suitable type of engine.

Eighth Division.

Calculation of the dimensions of the Cylinders.

Second Part.

Marine Boilers.

Ninth Division.

Types of Marine Boilers.

Tenth Division.

Strength of Marine Boilers.

Eleventh Division.

Furnaces.

Twelfth Division.

Marine Boiler Fittings.

Thirteenth Division.

The fitting of Boilers on board.

Errata.

for	read
Page 12, last line of 16), I II I	I_1
Page 15, 14 th line from bottom, λ_x	λ^x
Page 27, 4 th „ of 9), $r = 606.5 + 305\,t - (t + 0.00002\,t^2 + 0.0000003\,t^3)$	$r = 606.5 + 0.305\,t - (t + 0.00002\,t^2 + 0.000000\,t^3)$
Page 118, Eq. 89, $\eta_c = \frac{Q(T-T_1)}{L\,T}$	$\eta_c = \frac{Q(T-T_1)}{A\,L\,T}$
Page 164, Eq. 92[a], $e_a = 8100\,C + 29000\left(H + \frac{0}{8}\right) + 2500\,S - 600$ T. U.	$e_a = 8100\,C + 29000\left(H + \frac{O}{8}\right) + 2500\,S - 600\,W$ T. U.
and Eq. 92[b], $e_a = 8000\,C + 29000\left(H + \frac{0}{8}\right) + 2500\,S - 600$ T. U.	$e_a = 8000\,C + 29000\left(H + \frac{O}{8}\right) + 2500\,S - 600\,W$ T. U.
Page 200, 14th line from top, 320^0	325^0
and 15th line from top, $t_1 - t = 320 - 20 = 300^0$	$t_1 - t = 325 - 20 = 305^0$, say 300^0
Page 250, 7th and 8th line of 3), 1 HP Engl. $= 0.9863$ HP metr.	1 HP Engl. $= 1.0139$ HP metr.
1 HP metr. $= 1.0139$ HP Engl.	1 HP metr. $= 0.9863$ HP Engl.
Page 269, 7th line of 11), *L. P.* cylinder	*H. P.* cylinder
Page 322, Eq. 186, $W = 11\,\frac{\mathfrak{B}\,B}{\sqrt{B^2 - m\,L^2}}\,V^{2.5} - k\,S\,v^2$ kg	$W = 11\,\frac{\mathfrak{B}\,B}{\sqrt{B^2 + m\,L^2}}\,V^{2.5} - k\,S\,v^2$ kg.
Page 373, 11th line from bottom 3 kg	0.3 kg
and 8th line from bottom 1.5 kg	0.15 kg

First Division.

Principles of the mechanical Theory of Heat.

§ 1.

Fundamental Laws.

1) Every phenomenon in the Universe, which can be definitely explained, has its ultimate cause in the existence of Matter and Motion. **Fundamental Laws.**

2) *Matter* can neither be created nor destroyed, but only transformed. (Law of the conservation of Matter.) **Matter.**

3) Likewise *Energy* or *capacity for doing mechanical work*, which is the cause of all motion, can be neither produced nor extinguished, but only converted. (Law of the conservation of energy.) **Energy.**

§ 2.

Matter.

1) All bodies are composed of infinitely small particles which, in the chemical elements are called *Atoms* i. e. *indivisible*, and in chemical compounds *molecules* i. e. *small masses*. A molecule is composed of at least two atoms. **Atom and Molecule.**

2) The comparatively large spaces between the different molecules of a body are filled by a very attenuated and extremely elastic matter, *ether*, which pervades the entire Universe. The elasticity and density of this matter are determined by the **Ether.**

influence of the molecules of the body among which it is disposed.

Law of attraction.

3) The molecules of matter and the ether atoms attract each other, as is also the case with the molecules of matter themselves, but the ether atoms repel each other.

Dynamid.

4) In consequence of these forces an atmosphere of ether atoms is formed around each molecule of matter and such a molecule with its ether-atmosphere has been called (by Professor REDTENBACHER) a *Dynamid.**)

Bodies.

5) All bodies therefore are composed of a system of dynamids.

§ 3.

Energy.

Kinetic Energy.

1) Two free dynamids situated at any distance apart will attract or repel each other under the influence of the above forces until they reach their position of equilibrium. The work done during this motion is called *Kinetic energy, or energy of motion.*

Potential Energy.

2) If two dynamids are kept by force in a position in which, although subject to the influence of the before-mentioned forces, they cannot obey them, then the work stored up in the dynamids and available at any moment is called *potential energy, energy of position, or tension.*

Conversion of Energy.

3) When the two dynamids are in a state of motion (i. e. assuming the removal of the detaining force just referred to) their potential energy is gradually converted into kinetic energy, the sum of the two energies always remaining the same during the conversion; so that at the beginning of the motion with zero velocity and the potential energy at its original magnitude, the kinetic energy is also zero, whereas at the end of the motion and with the highest velocity, the potential energy is wholly converted into kinetic energy.

Oscillating or Vibratory motion of Dynamids.

4) In consequence of this kinetic energy, two dynamids in motion will approach each other within their distance of equilibrium, then repel each other beyond it, again approch &c., that is to say they will remain in *oscillating* or *vibratory* motion.

Oscillating motion of Ether atoms.

5) Two ether atoms of a dynamid behave towards each other exactly as the dynamids themselves, because each ether atom which is forcibly removed from its position of equilibrium, must for the above reasons assume an oscillating motion.

*) F. REDTENBACHER. Das Dynamidensystem. Mannheim 1857.

6) As in every case the molecule of a dynamid can revolve about its centre of gravity which is itself at rest when the molecule is in its position of equilibrium, it fallows that the atmosphere as well as the molecule of a dynamid may possess kinetic energy. **Motion of Molecules.**

7) The kinetic energy accumulated at any moment in a system of dynamids or in a body is called the *work of vibration* W of the body, from whatever sort of motions of ether atoms or molecules of matter it may arise. **Work of vibration.**

8) The potential energy contained in the same system of dynamids, as well as that portion of it which may have been used up in bringing about the momentary condition of the system is called the *work of disgregation* H. **Work of Disgregation.**

9) In taking effect, the work of disgregation has two classes of forces to overcome*): **Work of translation.**

   a) those which the particles of the body exert upon each other and are therefore characteristic of it,

   b) those which arise from external influences under which the body is placed and are generally evinced as pressures on its surface.

   The work necessary to overcome the forces (a) is called the *work of translation* I, and that necessary to overcome the forces (b) the *external work* L. **External Work.**

10) As the work of vibration W and the work of translation I only bring about internal changes of condition in a body, it is usual to include them in the term *internal work (or intrinsic energy)* U, as distinguished from the external work L. **Internal Work or Intrinsic Energy.**

11) As a system of dynamids can only experience a change of condition by a variation of its kinetic and potential energy, so a body can only suffer a change of state by receiving or parting with work of vibration and disgregation or both simultaneously, of whatever kind the influences may be to which it is subjected. **Changes of condition in bodies.**

§ 4.

Equivalence of Heat and Work.

First Law of Thermodynamics.

1) It is a matter of experience that a body exposed to the influences of heat, i. e. having heat imparted to or withdrawn from it, undergoes a corresponding increase or diminution of temperature and volume. **Effect of Heat.**

*) R. CLAUSIUS. "Die mechanische Wärmetheorie". Brunswick 1876. P. 29.

Sensible energy.

2) The change of temperature is attributed to variation in the work of oscillation, which is therefore called *sensible energy*. In accordance with this view the *sensible or measurable heat* in a body is a motion of its smallest particles. This highly probable inference is justified by a series of investigations which have demonstrated that *radiant heat*, like light, has its cause in the transmission of oscillating motions of ether.

Change of Volume.

3) Change of volume of a body being mostly accompanied by the overcoming of external resistance (pressure) leads in general to a change in the arrangement of the component parts of the body and can therefore only take place through the absorption or the giving out of work of disgregation.

Equivalence of Heat and Work.

4) Hence we assume *that the quantity of Heat which is imparted to or withdrawn from a body is directly proportional to the sum of the simultaneously occurring variations of the work of oscillation and disgregation.*

I Fundamental Law.

5) The correctness of the foregoing hypothesis has been shown by experiments in which a certain quantity of heat was produced by a certain quantity of work and vice versâ.

We therefore say

"*Heat and Work are equivalent*".

This is the first law of Thermodynamics.

Unit of Work.

6) The *unit of work* is the quantity of energy which must be exerted in order to raise one kilogram one metre high, it is called the *kilogrammetre*, *mk.*

Unit of Heat.

7) The *unit of heat* or thermal unit is the quantity of heat which must be communicated to a kilogram of water in order to raise its temperature from 0° C.*) to 1° C.

8) Extraordinarily careful and numerous experiments of JOULE**) and others have confirmed the probability of the relation

$$1 \text{ thermal unit} = 424 \text{ mk} = \frac{1}{A} \quad \ldots\ldots\ldots\ldots \quad (1)$$

Mechanical Equivalent of Heat.

This work of 424 mk is called the

mechanical equivalent of heat,

or we may write

$$\tfrac{1}{424} \text{ of a thermal unit} = 1 \text{ mk} = A \quad \ldots\ldots\ldots \quad (1^a)$$

and call A the *heat equivalent of the unit of work.*

Communication of Heat in general.

9) It may be inferred from the above that if a certain quantity of heat dQ is communicated to a body from a source of heat,

a) one part of this heat $A\,dW$ is used in increasing the work of oscillation, thus raising the temperature,

*) Whenever in this work degrees of temperature are used without further distinction, the Centigrade Scale is always referred to.

**) Philosophical transactions for the year 1850. P. 1.

b) another part $A\,d\,I$ changes the work of translation i. e. causes a variation of the body's cohesion,

c) the remaining part $A\,d\,L$ performs the external work necessary to overcome the pressure resisting the expansion of the body

$$\therefore\ d\,Q = A\,(d\,W + d\,I + d\,L)^{*}) \qquad (2)$$

or as $\quad d\,W + d\,I = d\,U$ and $d\,I + d\,L = d\,H$

$$d\,Q = A\,(d\,U + d\,L) = A\,(d\,W + d\,H) \qquad (2^{a})$$

If the quantity of heat $d\,Q$ is withdrawn from the body, the signs in these equations become negative.

10) CLAUSIUS**) describes the heat motion in the interior of bodies in different states of aggregation in the following manner: **Motion of Heat.**

a) In a *solid* the molecules oscillate in straight lines about certain positions of equilibrium as influenced by the forces they exert upon each other; revolving motions about their own centres of gravity, as well as motions of the atoms of each molecule can also arise.

b) In a *liquid* an oscillating, rolling, and progressive motion takes place, the kinetic energy of which is however not sufficient to separate the molecules by overcoming their mutual attraction; they therefore remain within a certain volume without external pressure.

c) In a *perfect gas* (see § 6, 1) the molecules move rectilineally according to the law of inertia, while revolving about their own centres of gravity. They are removed entirely beyond the range of their mutual attraction.

11) If heat is communicated to or withdrawn from a body, we have **Communication of Heat under particular circumstances.**

$$d\,W = 0$$

when the body's state of aggregation *alone* is changed,

$$d\,I = 0$$

when the body is a *perfect gas*,

$$d\,L = 0$$

when the body's *Volume remains constant.*

12) This scheme illustrates the changes arising in a body under the influence of heat **Scheme of the changes produced by Heat.**

Heat imparted or withdrawn $(d\,Q)$

Sensible Energy $(d\,W)$ — Work of Disgregration $(d\,H)$

Work of Translation $(d\,I)$ — External Work $(d\,L)$

Change of Temperature. — Change of Cohesion. — Change of Pressure.

Internal Work $(d\,U)$.

*) G. ZEUNER. Technische Thermodynamik. Leipzig 1887. P. 21.

**) POGGENDORFF'S Annalen, Vol. 100, P. 353.

§ 5.

Specific Heat.

Specific Heat for equal weights.

1) The quantity of heat i. e. the number of units of heat which must be imparted to one kilo of any kind of matter to raise its temperature 1^0 is called its *specific heat for equal weights.*

Specific Heat for equal volumes.

2) The quantity of heat i. e. the number of units of heat which must be imparted to one cubic metre of any kind of matter to raise its temperature 1^0 is called its *specific heat for equal volumes.*

Relation of the two specific Heats.

3) The specific heat for equal volumes of any substance equals its specific heat for equal weights multiplied by its specific gravity.

Capacity for Heat.

4) As only the specific heat for equal weights is usually in question, this is generally designated simply as the *heat capacity or specific heat C.*

Water value.

5) The *water value CG* of a substance is the product of its specific heat C and its weight G and represents the weight of water in which any quantity of heat will produce the same rise of temperature as in the weight G of the substance itself.

Variability of Specific Heat.

6) The foregoing definitions have come down to us from the time when it was believed that heat was itself a substance, and it was unknown that the specific heat of all bodies except the perfect gases is more or less a function of their temperature.*) For such technical calculations only as do not require a very high degree of accuracy,**) the specific heat C can be taken as constant for each substance. For instance in the case of water it has, according to REGNAULT, the following relation to the temperature

$$C = 1 + 0{,}0000\,4\,t + 0{,}000000\,9\,t^2.$$

The following table shews the specific heats of the most important substances for the engineer as determined by REGNAULT.

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*) According to G. ZEUNER. "Technische Thermodynamik". Leipzig 1887. P. 141 it appears that the specific heats of the gases possibly vary with the Temperature *and* the Pressure.

**) More exact data as to the variability of the specific heat with the temperature and state of aggregation are collated by PFAUNDLER in MÜLLER-POUILLET's Lehrbuch der Physik. Vol. II. Div. II. P. 318.

Table of specific Heats of Solids and Liquids.

Substance	Specific heat with equal weight	Specific heat with equal Volume	Substance	Specific heat with equal weight	Specific heat with equal Volume
1	2	3	1	2	3
Antimony	0,0508	340	Wrought iron	0,1138	882
Lead	0,0314	357	Steel, hard	0,1175	916
Cast iron	0,1298	941	Steel, soft	0,1165	916
Wood, Oak	0,5700	439	Zink	0,0955	688
Wood, Pine	0,6500	280	Tin	0,0562	410
Copper	0,0951	833	Mercury	0,0333	—
Brass	0,0939	803	Water	1,0000	1000

7) In the case of gases, the *specific heat under constant volume* C_v is distinguished from the *specific heat under constant pressure* C_p, according to whether they are heated under constant volume or constant pressure. Specific Heat of Gases.

8) If heat is imparted to 1 kilo of a perfect gas under constant volume it can only produce sensible energy, as dL and dI are zero, and if the temperature of the gas is raised 1^0, we have Specific heat under constant Volume Cv.

$$dQ = A\,dW = C_v$$

If the temperature of a constant volume of a perfect gas rises dT^0, the sensible energy produced is in general

$$A\,dW = C_v\,dT \quad \ldots\ldots\ldots\ldots\ldots \quad (3)$$

9) If the temperature of 1 kilo of a perfect gas is raised 1^0 under constant pressure the imparted heat is converted into sensible energy and external work only, because $dI = 0$. The sensible energy is by 8) $= C_v$ and the external work is in this case constant and $= R$, as is shewn in § 6, 18. From this follows Specific heat under constant Pressure Cp.

$$\left.\begin{array}{l} C_p = C_v + AR \\ C_p - C_v = AR \end{array}\right\} \quad \ldots\ldots\ldots\ldots\ldots \quad (4)$$

10) Finally, it has been shewn by experiment and calculation that the ratio $\frac{C_p}{C_v}$ for perfect gases is constant Ratio of the two specific Heats.

$$\frac{C_p}{C_v} = \varkappa = 1{,}41 \quad \ldots\ldots\ldots\ldots\ldots \quad (5)$$

From equation 4 and 5 it follows that

$$\frac{C_p}{C_v} = \frac{AR}{C_v} + 1 = \varkappa \quad \ldots\ldots\ldots\ldots\ldots \quad (5^a)$$

$$R = (\varkappa - 1)\frac{C_v}{A} \quad \ldots\ldots\ldots\ldots\ldots \quad (6)$$

$$AR = (\varkappa - 1)\,C_v = C_p - C_v \quad \ldots\ldots\ldots\ldots \quad (7)$$

11) The following table gives the specific heats of several gases and gaseous mixtures of the greatest practical importance

Gas	Density compared to air δ	Specific heat under constant pressure C_p	Specific heat under constant Volume C_v	$\varkappa = \frac{C_p}{C_v}$	Specific heat under constant pressure compared to the specific heat of air = 1
1	2	3	4	5	6
Atmospheric Air	1,0000	0,2375	0,1684	1,410	1,0000
Oxygen	1,1056	0,2175	0,1550	1,403	0,9158
Nitrogen	0,9714	0,2438	0,1727	1,412	1,0265
Hydrogen	0,0693	3,4090	2,4119	1,413	14,3537
Carbonic Oxide	0,9673	0,2450	0,1736	1,411	1,0316
Carbonic Acid at 0°	1,5291	0,1870			0,7874
„ „ „ 100°	—	0,2145	0,1714 mean	1,265 mean	0,9032
„ „ „ 200°	—	0,2396			1,0088
Marsh Gas	0,5527	0,5929	0,4679	1,267	2,4964
Olefiant Gas	0,9672	0,4040	0,3326	1,215	1,7011
Steam (moderately superheated)	0,6219	0,4804	0,3694	1,301	2,0232

§ 6.

Laws of the changes of state of perfect Gases.

Perfect Gases.

1) If a gas is so far removed from its point of condensation that the following laws are applicable with exactness to its changes of state, it is called a *perfect* Gas; for all other gases these laws are only approximately correct.

Diagrams of Energy.

2) The changes of state of a gas can be graphically represented by placing the successive volumes as abscissæ and the corresponding pressures as ordinates in a system of orthogonal coordinates. The ends of the ordinates for one and the same change of state will then form a curve (Plate 1, Fig. 3) the *curve of pressure* or *tension curve* and the tension curves of all changes of state which differ from each other enclose a space, which is called a *diagram of energy*. The area of it represents (to any particular scale) the external work produced or expended through or during the changes of state of the gas.

Gay-Lussac's Law.

3) **Gay-Lussac's Law:** *All perfect gases expand equally when heated under constant pressure.*

4) If a volume v_0 of a perfect gas under the atmospheric pressure (i. e. with the barometer at 760 mm) is heated from 0^0 to t^0, it expands, according to the researches of GAY-LUSSAC, REGNAULT and others, about $^1/_{273}\ v_0$ for each degree of rise of temperature. The number $^1/_{273} = \alpha = 0{,}003665$ is called the *coefficient of expansion of the perfect gases.* At the end of the heating process v_0 becomes increased to

Coefficient of Expansion.

$$v = v_0\,(1 + \alpha\, t) \quad \ldots\ldots\ldots\ldots\ldots \quad (8)$$

5) If the same volume v_0 be cooled to a lower temperature than 0^0, it must also decrease by $^1/_{273}\ v_0$ for each degree, so that it will reach the zero limit of its volume at -273^0. It is assumed that at this limit temperature ceases altogether to exist and the temperature of -273^0 is called the *absolute zero of temperature.*

Zero of Temperature.

6) The temperature of a body measured from this zero is called its *absolute temperature*, and this equals, if its temperature on the centigrade scale is called t

Absolute Temperature.

$$T = 273 + t \quad \ldots\ldots\ldots\ldots\ldots \quad (9)$$

7) From 4) and 6) we may derive

Gay-Lussac's Law expressed in another way.

$$\frac{v}{v_1} = \frac{v_0\,(1 + \alpha\, t)}{v_0\,(1 + \alpha\, t_1)} = \frac{\frac{1}{\alpha} + t}{\frac{1}{\alpha} + t_1} = \frac{273 + t}{273 + t_1} = \frac{T}{T_1} \quad \ldots\ldots \quad (10)$$

that is to say

The Volumes of all perfect gases are, under constant pressure, directly proportional to their absolute temperatures.

8) *The external work* which a perfect gas performs while expanding according to GAY-LUSSAC's law from v to v_1 in consequence of its temperature being raised from T^0 to $T_1{}^0$ is calculated from Eq. 18 (see below, 19)

External work under Gay-Lussac's Law.

$$d\,L = p\,d\,v$$

or as p is constant

$$\int d\,L = p \int_v^{v_1} d\,v$$

$$L = p\,(v_1 - v) \text{ kilogrammetres (mk)} \quad \ldots\ldots\ldots \quad (11)$$

9) *The quantity of heat*, which must be imparted to every kilo of the gas in order that it may perform the above work in accordance with GAY-LUSSAC's law, is determined from the general equation

Quantity of heat under Gay-Lussac's Law.

$$d\,Q = A\,(d\,W + d\,I + d\,L).$$

In all perfect gases $d\,I = 0$

$$d\,Q = A\,(d\,W + d\,L)$$

and as, according to Eq. 3

$$A\,d\,W = C_v\,d\,T$$

$$A\int d\,W = C_v\int_T^{T_1} d\,T$$

$$A\,W = C_v\,(T_1 - T)$$

it follows that

$$Q = A\,W + A\,L = C_v\,(T_1 - T) + Ap\,(v_1 - v).$$

remembering that p is constant, by Eq. 16 we have

$$T_1 = \frac{pv_1}{R} \text{ and } T = \frac{pv}{R}$$

so that

$$Q = \left(\frac{C_v}{R} + A\right) p\,(v_1 - v) = \left(\frac{C_v}{AR} + 1\right) Ap\,(v_1 - v)$$

from this follows by Eq. 7

$$Q = \frac{\varkappa}{\varkappa - 1} Ap\,(v_1 - v) \quad \text{units of heat} \quad \ldots\ldots (12)$$

Example of Gay-Lussac's Law.

10) Example: 1 kilo of a perfect gas, the constant R of which is taken at 30 (see below 20) possesses an absolute temperature $T = 300^0$ and is under a pressure $p = 40000$ kilos per sq. cm. Its volume v by Eq. 16 is then

$$v = \frac{R\,T}{p} = \frac{30 \times 300}{40000} = 0{,}225 \text{ cubic metres.}$$

Let this gas expand according to GAY-LUSSAC's law under *constant pressure* to a volume

$$v_1 = 4\,v = 4 \times 0{,}225 = 0{,}9 \text{ cubic metres}$$

it will then perform the external work (work of expansion)

$$L = p\,(v_1 - v) = 40000\,(0{,}9 - 0{,}225) = 27000 \text{ kilogrammetres.}$$

The quantity of heat which must be imparted to it during the expansion is

$$Q = \frac{\varkappa}{\varkappa - 1} Ap\,(v_1 - v) = \frac{1{,}41}{1{,}41 - 1} \times \frac{1}{424} \times 27000 = 219{,}06 \text{ units of heat}$$

of which $\frac{27000}{424} = 63{,}68$ are expended in the work of expansion and $219{,}06 - 63{,}68 = 155{,}38$ in the rise of temperature.

Gay-Lussac's Curve.

11) The tension curve for GAY-LUSSAC's law is a straight line parallel to the axis of abscissæ, because the pressure for the successive volumes of the gas is constant. It is shewn in Fig. 2, Plate 1 as the straight line $A\,0$ produced to the right past 0.

Mariotte's Law.

12) **Mariotte's Law:** *The volumes of all perfect gases at constant temperatures are inversely proportional to their pressures.*

$$\left.\begin{aligned} \frac{v}{v_1} &= \frac{p_1}{p} \\ p\,v &= p_1\,v_1 \end{aligned}\right\} \ldots\ldots\ldots\ldots (13)$$

13) *The external work* which a perfect gas performs in expanding under a constant temperature T^0 from v to v_1 under MARIOTTE's law, is determined in the following manner by the help of Eq. 18 **External work under Mariotte's Law.**

$$d L = p \, dv$$

in which p and v are variables. To eliminate p we put, by Eq. 16

$$p = \frac{R T}{v}$$

then

$$d L = R T \frac{dv}{v}$$

$$\int d L = R T \int_v^{v_1} \frac{dv}{v}$$

$$L = R T \log. \text{nat.}^{*)} \frac{v_1}{v} = R T \log. \text{nat.} \frac{p}{p_1} \quad \text{kilogrammetres} \; . \; . \; (14)$$

14) *The quantity of heat* which must be communicated to each kilogram of the gas in order that it may perform this work, is calculated as follows from the general equation **Quantity of heat under Mariotte's Law.**

$$d Q = A (d W + d I + d L)$$

in this case $d I = 0$ because it refers to a perfect gas
$d W = 0$ because the temperature is constant

then $d Q = A \, d L$

$Q = A L$

or referring to Eq. 14

$$Q = A R T \log. \text{nat.} \frac{v_1}{v} \quad \text{units of heat} \; . \; . \; . \; . \; . \; . \; . \; (15)$$

15) Example: If the kilo of gas referred to in 10) expands according to MARIOTTE's Law at *constant temperature*, the work of expansion produced by it will be **Example of Mariotte's Law.**

$$L = R T \log. \text{nat.} \frac{v_1}{v} = 30 \, . \, 300 \, . \log. \text{nat.} \, 4 = 12476 \text{ kilogrammetres}$$

and the quantity of heat communicated to it

$$Q = A L = \frac{1}{424} \, . \, 12476 = 29{,}42 \text{ units of heat.}$$

16) The expansion curve drawn according to MARIOTTE's law is called an *Isothermal;* it is an equilateral hyperbola whose asymptotes are the axes of coordinates. It may be drawn according to RANKINE**) for an initial volume v expanding to v_1, by dividing the distance $v_1 - v$ (Plate 1, Fig. 1) into any number of equal or unequal parts, erecting perpendiculars from the points so obtained and then drawing a parallel to $v_1 - v$ through the end of the ordinate which represents the initial **Mariotte's Curve or Isothermal.**

*) log. nat. = *logarithmus naturalis*, the natural logarithm.

**) W. J. M. RANKINE. Miscellaneous Scientific Papers. London 1881. P. 456.

pressure p. Join the points of intersection 1, 2, 3 . . . of this parallel and the perpendiculars, with the origin O of the coordinates. Through the intersections I, II, III . . ., of these oblique lines and the ordinate p lay parallels to $v_1 - v$. Then the intersections 1_I; 2_{II}; 3_{III} . . ., of these parallels with the respective ordinates will be points of the required Isothermal. The proof of this construction lies in the similarity of the corresponding triangles O A 1 and I II 1.

Combined Law of Gay-Lussac and Mariotte.

17) **The combined law of Gay-Lussac and Mariotte:** *In all perfect gases the external work, produced by heating 1 kilo 1° under constant pressure, is constant.*

Derivation of the Law.

18) According to Eq. 8 let a kilo of a perfect gas at t^0 under the atmospheric pressure have a volume $v = v_0\ (1 + \alpha t)$; at a pressure p its volume then becomes according to Eq. 13

$$v_1 = \frac{v_0}{p}\,(1 + \alpha t)$$

and the volume v_1 of 1 kilo of a perfect gas at $t^0{}_1$ and a pressure p_1.

$$v_1 = \frac{v_0}{p_1}\,(1 + \alpha t_1) \qquad \text{so that}$$

$$\frac{v}{v_1} = \frac{p_1}{p}\,\frac{(1 + \alpha t)}{(1 + \alpha t_1)} = \frac{p_1 \left(\frac{1}{\alpha} + t\right)}{p\left(\frac{1}{\alpha} + t_1\right)} = \frac{p_1}{p}\,\frac{T}{T_1}$$

$$\frac{p\,v}{T} = \frac{p_1\,v_1}{T_1} = R = \text{const.} \qquad (16)$$

$$p\,v = R\,T \qquad (17)$$

Equation of state of the perfect gases.

19) Eq. 17 is called the *equation of state of the perfect gases;* it expresses the law that when 1 kilo of a perfect gas is heated from the absolute zero to T^0 under constant pressure p it produces an external work of $p\,v$ kilogrammetres or $\frac{p\,v}{T} = R$ kilogrammetres per degree. The infinitely small increment of the external work amounts therefore to

$$d\,L = p\,d\,v \qquad (18)$$

Constant of the Law of Gay-Lussac and Mariotte.

20) R is called the *constant of the law of Gay-Lussac and Mariotte.* $R = 29{,}272$ for pure and dry atmospheric air, the density of which at $t = 0^0$ with the barometer at 760 mm is $0{,}0012932 = \frac{1}{773}$ (the density of water being 1). For other gases or gaseous mixtures, whose density is δ referred to the density of the air at the same temperature and pressure (see table on P. 8)

$$R = \frac{29{,}272}{\delta}$$

According to REGNAULT, for

Nitrogen	$R = 30{,}134$
Oxygen	$R = 26{,}475$
Hydrogen	$R = 422{,}612$
Carbonic acid at 0^0	$R = 19{,}143$
Steam at atmospheric pressure	$R = 47{,}023$

21) **Poisson's Law:** *The volumes of all perfect gases in nonconducting vessels are inversely proportional to the $\varkappa^{th}$ ($1{,}41^{th}$) roots of their pressures and to the $(\varkappa - 1)^{th}$ ($0{,}41^{th}$) roots of their absolute temperatures.* Poisson's Law.

22) In a nonconducting vessel heat is neither imparted to nor withdrawn from a gas during its change of state. Nonconducting Vessels.

23) In such a change of state therefore, we have according to Eq. 2 $\quad dQ = A(dW + dI + dL) = 0$ Relations between Pressure, volume, and temperature of perfect gases under Poisson's law.
in perfect gases $dI = 0$; further according to Eq. 3: $A\,dW = C_v\,dT$ and according to Eq. 18, $dL = p\,dv$

$$\therefore\ dQ = C_v\,dT + A p\,dv = 0$$

From this expression we may derive the following relations

between volume and pressure of the gas.

Differentiating Eq. 17, in order to eliminate T, we get

$$dT = \frac{1}{R}(p\,dv + v\,dp)$$

and then by substituting this value

$$\frac{C_v}{R}\,v\,dp + \left(\frac{C_v}{R} + A\right)p\,dv = 0$$

Referring to Eq. 5ᵃ, we may put for this

$$\frac{C_v}{R}(v\,dp + \varkappa p\,dv) = 0$$

Multiplying this Equation by $\frac{R}{C_v p v}$, we get

$$\frac{dp}{p} + \varkappa\frac{dv}{v} = 0;\ \frac{dp}{p} = -\varkappa\frac{dv}{v}$$

$$\int_p^{p_1}\frac{dp}{p} = -\varkappa\int_v^{v_1}\frac{dv}{v}$$

$$\log\text{nat}\,(p_1 - p) = -\varkappa\log\text{nat}\,(v_1 - v)$$

$$\log\text{nat}\,\frac{p_1}{p} = \varkappa\log\text{nat}\,\frac{v}{v_1}$$

$$\frac{p_1}{p} = \left(\frac{v}{v_1}\right)^{\varkappa}$$

between volume and temperature of the gas.

To eliminate p, we put, by Eq. 16

$$p = \frac{RT}{v}$$

then

$$C_v\,dT + ART\frac{dv}{v} = 0$$

Referring to Eq. 7, this Equation may be expressed as

$$C_v\left[dT + (\varkappa - 1)\,T\frac{dv}{v}\right] = 0$$

Dividing this Equation by C_v we get

$$dT + (\varkappa - 1)T\frac{dv}{v} = 0;\ dT = -(\varkappa - 1)T\frac{dv}{v}$$

$$\int_T^{T_1}\frac{dT}{T} = -(\varkappa - 1)\int_v^{v_1}\frac{dv}{v}$$

$$\log\text{nat}\,(T_1 - T) = -(\varkappa - 1)\log\text{nat}\,(v_1 - v)$$

$$\log\text{nat}\,\frac{T_1}{T} = (\varkappa - 1)\log\text{nat}\,\frac{v}{v_1}$$

$$\frac{T_1}{T} = \left(\frac{v}{v_1}\right)^{\varkappa - 1}$$

Combining both these equations we get

$$\left.\begin{aligned} \frac{v}{v_1} &= \left(\frac{p_1}{p}\right)^{\frac{1}{\varkappa}} = \left(\frac{T_1}{T}\right)^{\frac{1}{\varkappa-1}} \\ \frac{p}{p_1} &= \left(\frac{v_1}{v}\right)^{\varkappa} = \left(\frac{T}{T_1}\right)^{\frac{\varkappa}{\varkappa-1}} \\ \frac{T}{T_1} &= \left(\frac{v_1}{v}\right)^{\varkappa-1} = \left(\frac{p}{p_1}\right)^{\frac{\varkappa-1}{\varkappa}} \end{aligned}\right\} \quad \ldots\ldots \quad \ldots . \tag{19}$$

External work under Poisson's Law.

24) *The external work*, which 1 kilo of a perfect gas performs in expanding under Poisson's law from v to v_1, while its temperature sinks from T^0 to $T_1^{\,0}$ is calculated from the general equation

$$d\,Q = A\,(d\,W + d\,I + d\,L)$$

in which we have

$d\,Q = 0$, because heat is neither imparted to nor withdrawn from the gas, and

$d\,I = 0$, because it is a perfect gas.

Therefore we have remaining

$$A\,(d\,W + d\,L) = 0;\ A\,d\,W = -A\,d\,L$$

$$A\,d\,W = C_v\,d\,T$$

$$A\int d\,W = C_v\int_T^{T_1} d\,T = -A\int d\,L$$

$$C_v\,(T_1 - T) = -A\,L$$

$$L = \frac{C_v}{A}(T - T_1)$$

Now by Eq. 19

$$T_1 = T\left(\frac{v}{v_1}\right)^{\varkappa-1} = T\left(\frac{p_1}{p}\right)^{\frac{\varkappa-1}{\varkappa}}$$

$$\therefore L = \frac{C_v}{A}\left[T - T\left(\frac{v}{v_1}\right)^{\varkappa-1}\right] = \frac{C_v}{A}\,T\left[1 - \left(\frac{v}{v_1}\right)^{\varkappa-1}\right]$$

If in this expression, we substitute by Eq. 17 $T = \frac{p\,v}{R}$ and by Eq. 7 $\frac{C_v}{AR} = \frac{\varkappa - 1}{1}$, it follows that

$$L = \frac{p\,v}{\varkappa - 1}\left[1 - \left(\frac{v}{v_1}\right)^{\varkappa-1}\right] = \frac{p\,v}{\varkappa - 1}\left[1 - \left(\frac{p_1}{p}\right)^{\frac{\varkappa-1}{\varkappa}}\right] \text{ kilogrammetres} \tag{20}$$

25) Example: the kilo of perfect gas referred to in 10) in expanding under Poisson's law from v to $v_1 = 4v$, performs the work

$$L = \frac{40000 \,.\, 0,225}{1,41 - 1} \left[1 - \left(\frac{0,225}{0,9} \right)^{1,41 - 1} \right]$$

$$L = \frac{9000}{0,41} \left(1 - 0,25^{0,41} \right) = 9519,61 \text{ kilogrammetres.}$$

To produce this work we require to expend a quantity of heat

$$Q = C_v (T - T_1) = A L = \frac{1}{424} \,.\, 9519,61 = 22,45 \text{ units of heat.}$$

26) The expansion curve drawn according to Poisson's law is called the *Adiabatic Curve* and is similar to the Isothermal Curve. Poisson's Curve or Adiabatic Curve.
BRAUER*) gives the following method of plotting it: it is evident from the equation $\frac{p_1}{p} = \left(\frac{v}{v_1}\right)^\varkappa$ of the curve that if we take two spots $(p\, v)$ and $(p_1\, v_1)$ so that $\frac{v}{v_1}$ has any value we choose, there is a corresponding value of $\frac{p_1}{p}$.

Thence it follows that for a series of abscissæ v, v_1, v_2, v_3, in geometrical progression, the ordinates p, p_1, p_2, p_3, must also form a geometrical series.

For instance if

$\lambda = \frac{v}{v_1} = \frac{v_1}{v_2} = \frac{v_2}{v_3} \ldots.$ is the factor of the series for v we shall get,

$\lambda^\varkappa = \frac{p_1}{p} = \frac{p_2}{p_1} = \frac{p_3}{p_2} \ldots$ the factor of the series for p.

In Plate I Fig. 2 let O be the given point with the coordinates v and p through which the adiabatic curve is to be drawn. Choose a value for v_1, e. g. $v_1 = {}^5/_4\, v$ (in the Fig. $v = O a$, $v_1 = O b$, the Volumes V, V_1 and V_2 do not refer to this construction but to § 6, 29 and 30), through the terminal point b draw a line at 45°, cutting $O a$ produced in the point h and through O and h draw the straight line $O h$, by means of which the construction $a\, h\, b\, i\, c\, k\, l$ can be completed and the abscissæ cut off in geometrical progression. Now

$$p_1 = ({}^4/_5)^\varkappa\, p$$

is calculated, substituting the known value of $\varkappa$ and thus the point 1 is found. Then the point H in $1\,B$ is determined by the intersection of the line $A H$ drawn at 45°, and by joining

*) Zeitschrift des Vereines deutscher Ingenieure. 1885. P. 433.

this point H with O we get the construction line OH. How the other ordinates are obtained can be at once seen from the drawing. In this case $\varkappa$ is taken $= 1{,}41$, so that

$$p_1 = (4/5)^{1,41}\, p = 0{,}730\, p.$$

By exactly the same construction the curve can also be continued backwards from O. The points -1 and -2 are so found. For $\varkappa = 1$ the curve becomes the equilateral hyperbola or isothermal curve (shewn by the dotted line) so that the same construction can also be used for that curve.

A purely graphical method of laying down the adiabatic curve, which also avoids the above small logarithmic calculation, is given by TAUBELES*), but as this necessitates a more complicated construction and for many values of $\varkappa$ only furnishes an approximation to the adiabatic curve, it is not further described here.

Changes of state under constant volume.

27) **Changes of state of less importance** occur with permanent gases, when

a) *the volume remains constant*, i. e. when the gas is heated under a constant volume. In this case we have by Eq. 16

$$\frac{pv}{T} = \frac{p_1 v}{T_1};\ \frac{p}{p_1} = \frac{T}{T_1}.$$

Further, the external work $L = 0$, so that the quantity of heat Q imparted to the 1 kilo of perfect gas only produces the sensible energy W, the relation of which to Q is worked out in 9) as

$$Q = AW = C_v\,(T_1 - T) \text{ units of heat.}$$

For the gas as taken in 10) the quantity of heat necessary to produce this effect, as there calculated, is

$$Q = AW = 155.38 \text{ units of heat.}$$

Change of state with the internal work constant.

b) *The internal work U remains constant.* For perfect gases, this change of state is the same as that in which the temperature T as well as the sensible energy W are constant, i. e. the change occurs in accordance with MARIOTTE's law, because with perfect gases the work of translation $J = 0$. For any other gases or vapours this is not the case, *their* curve of tension with sensible energy and work of translation constant, is called the isodynamic curve.

Comparison of the different changes of state.

28) The difference (for 1 kilo of a perfect gas) between the quantity of heat imparted, Q, and the external work produced L, during the above described changes of state, can be best seen by placing side by side the respective tension-curves as in Fig. 2 Plate 1 or the calculated results as below:

*) Zeitschrift d. V. d. I. 1885. P. 675.

a) v const., p & T variable	: $Q = 155.33$ units of heat;	$L = 0.0$ kgmetres;	heated with constant Volume.
b) p const., v & T „	: $Q = 209.06$ „ „ „ ;	$L = 27000.0$ „ ;	Gay-Lussac's Law.
c) T const., p & v „	: $Q = 29.42$ „ „ „ ;	$L = 12476.0$ „ ;	Mariotte's Law.
d) U const., p & v „	: $Q = 29.42$ „ „ „ ;	$L = 12476.0$ „ ;	Isodynamic Change.
e) $Q = 0, p, v$ & T „	: $Q = 0.0$ „ „ „ ;	$L = 9520.0$ „ ;	Poisson's Law.

29) The isothermal and adiabatic changes of state, that is those which take place under MARIOTTE's and POISSON's Laws respectively, are the most important in engineering. It follows from the above comparative table that one kilo of a perfect gas, in expanding to four times its initial volume, performs $\frac{12476}{5919.61} = 1.31$ times as much work under isothermal as under adiabatic expansion. Further, on looking at Fig. 2 Plate 1, it is seen that a perfect gas of the initial volume V and pressure p, in expanding, first isothermally and then adiabatically, to the volume V_1 has in the former case a greater terminal pressure p_{1v} than in the second case, where it is only p_{1a}. The reason of this is that in isothermal expansion the temperature is constant and therefore heat must be imparted to the gas, whereas adiabatic expansion takes place without the communication of any heat. Comparison of the isothermal and adiabatic work of expansion.

30) If, on the other hand, the same gas is compressed isothermally to one fourth of its original volume, only $\frac{1}{1.31} = 0.763$ times as much energy is required to accomplish this as would be necessary for adiabatic compression. As Fig. 2 Plate 1 shews, for the compression from V to V_2, the terminal pressure p_{2i} of the isothermal is less than that of the adiabatic p_{2a} because in the former case heat must be withdrawn, to keep the temperature of the gas constant, whereas the adiabatic compression is unaccompanied by withdrawal of heat. From this follows — Comparison of the isothermal and adiabatic work of compression.

31) A perfect gas, and atmospheric air is usually considered as one by engineers, performs the more work by expansion, the more heat we impart to it during the process. Without this additional heat the temperature of the gas sinks very considerably during expansion, for (e. g.) if one kilo. of air at 17° dilates to five times its original volume, the terminal temperature becomes by Eq. 19: Practical application of the above considerations.

$$T_1 = (273 + 17)\left(\frac{1}{5}\right)^{0.41} = 150^{\circ}$$

$$\therefore\ t_1 = 150 - 273 = -123^{\circ}.$$

This circumstance is made use of for the production of cold in freezing machines. The same gas can be compressed with so much the less expenditure of work the less its temperature

rises, hence it is profitable to construct air compressors, such as are used for charging the air-chambers of Whitehead Torpedoes, in such a manner that the heat arising from the compression may be withdrawn from the air *as rapidly as possible.*

§ 7.

The equivalence of conversions.*)

II. Law of Thermodynamics.

Reversible cycles.

1) If a perfect gas undergoes a series of changes of state, during every moment of which the pressure and temperature of all its interior particles are the same as those at its surface, and at the conclusion of these changes resumes its original condition, it is said to *pass through a cycle.* A cycle is called *reversible* when the original state of the gas is produced by repeating the whole of the series of changes of state, but in the reverse order. The conversion of heat into work of expansion or the conversion of work of compression into heat are changes which may produce reversible cycles.

Non-reversible cycles.

2) *A non-reversible cycle* occurs, when in the course of it, changes arise in which heat passes directly from a body of higher temperature to a body of lower temperature by radiation or convection, or in which heat is produced by the work of friction.

Carnot's cycle.

3) A cycle, in which a kilo of perfect gas undergoes only four changes of state, two isothermal and two adiabatic, as explained below, is called *a simple reversible cycle or Carnot's cycle.*

Process of Carnot's cycle.

4) In CARNOT's cycle (Fig. 3 Plate 1) one kilo of a perfect gas is allowed to expand
 a) isothermally from the volume v to v_1 the temperature T being of course kept constant by imparting a quantity of heat Q to the gas,
 b) then adiabatically from v_1 to v_2 while the temperature falls from T to T_1 heat being neither imparted nor withdrawn,
 c) Hereupon the gas is compressed isothermally from v_2 to v_3, during which compression the withdrawal of a quantity of heat Q_2 keeps the temperature T_1 constant,

*) R. CLAUSIUS. Die Mechanische Wärmetheorie. Brunswick 1876. P. 95.

d) and the process completed by further (adiabatically) compressing the volume v_3 back to the original volume v, while T_1 rises to T, as no addition nor withdrawal of heat takes place.

5) A diagram of energy drawn in accordance with CARNOT's cycle shews that during the process a certain external work L has been performed which is equal to the enclosed area F (see § 6, 2). This work can only be produced in consequence of the heat imparted Q being greater than that withdrawn Q_1. **Diagram of Energy for Carnot's cycle.**

Accordingly AL must equal $Q - Q_1$.

If the process be carried out in the reverse order, the same amount of work will be consumed and an equal quantity of heat produced.

6) *The energy produced or expended in a Carnot's cycle* is calculated as follows: **External Work of Carnot's cycle.**

according to Eq. 15 the quantity of heat imparted during the isothermal expansion is

$$Q = A\,R\,T \ln \frac{v_1}{v}$$

and the quantity withdrawn, $Q_1 = A\,R\,T_1 \ln \dfrac{v_2}{v_3}$

thence follows $Q : Q_1 = T \ln \dfrac{v_1}{v} : T_1 \ln \dfrac{v_2}{v_3}$

During the adiabatic change, the following relation exists, by Eq. 19, between the absolute temperature and the volume:

$$\frac{v_2}{v_1} = \left(\frac{T}{T_1}\right)^{\frac{1}{\varkappa - 1}} \quad \text{and} \quad \frac{v_3}{v} = \left(\frac{T}{T_1}\right)^{\frac{1}{\varkappa - 1}}$$

hence $\dfrac{v_2}{v_1} = \dfrac{v_3}{v}$ or $\dfrac{v_1}{v} = \dfrac{v_2}{v_3}$,

so that we get this ratio $\dfrac{Q}{Q_1} = \dfrac{T}{T_1}$ and we have: $Q_1 = \dfrac{Q\,T_1}{T}$

$$\therefore\ A\,L = Q - \frac{Q\,T_1}{T}$$

$$\left.\begin{aligned} A\,L &= \frac{Q}{T}(T - T_1) \quad \text{units of heat} \\ L &= \frac{Q}{A\,T}(T - T_1) \quad \text{kilogrammetres} \end{aligned}\right\} \dots\dots \ (21)$$

7) From the above we see that during a CARNOT's cycle two conversions occur; — first heat is converted into work (or vice versâ) and secondly heat of a higher temperature is changed to heat of a lower temperature (or vice versâ). In other words: if a quantity of heat is to be converted into work, it is necessary **Inference from Carnot's cycle.**

that another quantity of heat be simultaneously brought from a higher to a lower temperature.

II. Law of thermo-dynamics.

8) *Two conversions which exactly make up for, or balance each other (as in Carnot's cycle), without requiring any further permanent change to be made, are called equivalent.*

This is the second law of thermodynamics.

Equivalent Values.

9) The production from work or energy of a quantity of heat Q of temperature T has the equivalent value $\frac{Q}{T}$, the transition of a quantity of heat Q from a higher temperature T to a lower temperature T_1 has the equivalent value $Q\left(\frac{1}{T}-\frac{1}{T_1}\right)$.

Entropy or Heatweight and Head of Temperature.

10) ZEUNER calls the value $\frac{Q}{T}$ the *Heat-weight*, CLAUSIUS however calls it the conversion value or *Entropy**). The difference $T - T_1$ is designated by ZEUNER the *Head of Temperature.*

Another view of the II. Law.

11) We may convert the equation $\frac{Q}{Q_1}=\frac{T}{T_1}$

into $$\frac{Q}{T}=\frac{Q_1}{T_1}\,.$$

As Q represents the imparted or positive quantity of heat and Q_1 the negative or withdrawn one, we may also write

$$\frac{Q}{T}+\frac{Q_1}{T_1}=0 \quad \ldots\ldots\ldots\ldots\ldots \quad (22)$$

whence it follows that

"*In any Carnot's cycle the imparted and withdrawn heatweights or conversion values (or entropies) are equal to each other, or in other words their algebraic sum equals zero*".

External work greatest in Carnot's cycle.

12) ZEUNER has expressed Eq. 21 in the following manner: "*the quantity of heat converted into external work in a Carnot's cycle is equal to the heatweight (or entropy) multiplied by the head of temperature*". He proves besides *that**) of all similar cycles, Carnot's produces the greatest external work.*

Limits of Temperature fixed.

13) The external work increases with the head of temperature, which has a fixed value in many CARNOT's cycles, because the initial temperature T and the terminal temperature T_1 are confined within certain impassable limits. But as these temperatures are directly proportional to Q and Q_1 the heatweights imparted

*) R. CLAUSIUS. Die mechanische Wärmetheorie. Brunswick 1876. Pp. 111 and 204, also MÜLLER-POUILLET's Lehrbuch der Physik Vol. II, Div. II, P. 468. enlarged by PFAUNDLER. Brunswick 1879.

**) G. ZEUNER. Technische Thermodynamic. Leipzig 1887. P. 50.

and withdrawn, we arrive at the following important theorem (particularly so for the theory of the steam-engine).*)

14) *The external work which it is possible to obtain from a Carnot's cycle is a maximum when the whole communication of heat takes place at the highest temperature of the cycle, and the whole withdrawal at the lowest.* Perfect cycle.

Such a CARNOT's cycle is called a *perfect cycle.*

*) H. v. REICHE. Der Dampfmaschinen-Constructeur. Aachen 1880. P. 20.

Second Division.

Steam.

§ 8.

Different states of steam.

Vapour.

1) By *vapour* is meant a kind of gas which can be formed from a fluid by communicating heat or lowering pressure and which can on the other hand be made fluid by withdrawing heat or raising pressure.

Steam.

2) *Steam* occurs in two states of temperature, as
   I. saturated steam,
   II. superheated steam.

Saturated Steam.

3) **I. Saturated Steam** can only exist in contact with the fluid from which it was produced, although the quantity of this fluid may be very small. Its condition is a limiting one with respect to the transition to the fluid state of aggregation, because the slightest cooling entails a partial condensation. Saturated steam possesses for every pressure the lowest temperature, the smallest specific volume, and the greatest specific gravity; in other words it is always at its maximum density. Its temperature corresponding to any pressure is called *the temperature of saturation.*

Classification of steam.

4) With regard to saturated steam, a distinction is made in practice between
   a) dry steam,
   b) moist steam,
   c) wet steam.

Dry steam.

5) a. *Dry steam* is steam, the last fluid particle of which has just been evaporated, it is in fact exactly at its point of condensation. Dry steam can hardly be obtained except in superheaters, the purpose of which is, now-a-days, the production of the driest possible steam. — When in the following pages

"*saturated steam*" simply is spoken of, "*dry steam*" is always meant.

6) b. *Moist steam* is a mixture of steam and water, regarding which we do not yet know with certainty whether the water or moisture it contains floats like fine cloud in the mass of steam or precipitates itself in the form of dew on the walls of the containing vessel. Steam, on entering the cylinders of steam-engines, is usually moist; even when dried in superheaters, it again becomes moist through losses of heat occurring before the cut-off. But here we may at once remark that a degree of moisture of 10 to 20 $^0/_0$ at the boiler, as formerly assumed, does not occur. According to recent investigations, (see § 12) a *good* boiler, not over-driven, delivers steam which is either completely dry or only charged with a very small quantity (at most 5 $^0/_0$) of water carried over. **Moist steam.**

7) c. *Wet steam* contains so much water that this is injuriously noticeable in the cylinders, as in the case of priming or other mechanical carrying over of the water. **Wet steam.**

8) **II. Superheated Steam** can no longer exist when brought into contact with the fluid from which it originated. It has a higher temperature than saturated steam of equal pressure, as well as a greater specific volume and less specific gravity. Very highly superheated steam behaves like a perfect gas (see § 6, 1). **Superheated steam.**

9) In practice, a mixture of superheated with saturated steam is called *mixed steam* and is produced before the steam is admitted to the cylinder in order to keep too highly superheated steam away from the engine, for reasons to be explained further on (see § 13, 18). As the watery particles of the saturated steam are evaporated by the higher temperature of the superheated steam, the resulting *mixed steam* is either *superheated* steam of a temperature a little above that of saturation, or *moist* steam of a low degree of moisture, according to whether the above evaporation is complete or not. **Mixed steam.**

§ 9.

Definitions.

1) *The pressure* in kilos exerted by steam on every □cm of the walls of its containing vessel is called its *specific pressure*, *pressure*, or *expansive force p*. **Pressure p.**

Absolute pressure and "overpressure".

2) If the steam pressure is measured from a vacuum, i. e. from zero, it is called *absolute pressure*, but if measured from the pressure of the atmosphere it is called "*overpressure*" (or simply *pressure*). This latter pressure is shewn on steam-gauges.

Atmospheric pressure.

3) The atmospheric pressure which, at the mean barometric height, balances a column of mercury 76 cm high, amounts, for every □cm of the surface of the sea to 1.0334 kilos, so that

1 Atm. = 1.0334 kilos per □cm.

Notation of steam pressure.

4) Steam pressure can be expressed either in atmospheres or in kilos per □cm. Wherever the former is (exceptionally) the case in this book p bears the signification p *old* Atm. This distinction has become necessary since in the German Empire "*by atmospheric pressure, the pressure of 1 kilo per □cm*" is understood by law (§ 11, Section 2 "General Regulations as to the fitting up of boilers").*) In order to comply with the law, p is given in the following pages "*in kilos per □cm absolute pressure*", or as it is sometimes expressed "in new atmospheres of absolute pressure" and means p kilos per □cm or p atmospheres. In a few places where the steamgauge pressure is referred to, it is accentuated by being marked "*overpressure*". In reference to boilers and in conformity with practice, the term *Working Pressure* or *Boiler Pressure* is made use of, as signifying the legal pressure of a boiler and synonymous with *overpressure.*

Computation of the pressure p from the temperature t.

5) The pressure p of saturated, as well as moist steam or of a mixture of steam and water is independent of the volume and the degree of moisture or wetness and is to be regarded as a function of the temperature t only. The form of this function has not yet been theoretically determined, and REGNAULT gives the following empirical formula for it, based upon his experiments:

$$^{**)} \log p = a + b\,\alpha^{\tau} + c\,\beta^{\tau} \qquad \ldots\ldots\ldots (23)$$

in this expression are to be substituted

p the pressure in millimetres of mercurial column

$a = 4.7393707$ . . . . for $t = 0^0$ to 100^0

$a = 6.2640348$ . . . . for $t = 100^0$ to 200^0

$+ b\,\alpha^{\tau} = -\,num.\log(0.6117408 - 0.003274463\,t)$ . . for $t = 0^0$ to 100^0

$+ b\,\alpha^{\tau} = -\,num.\log(0.6593123 - 0.001656138\,t)$ . . for $t = 100^0$ to 200^0

$+ c\,\beta^{\tau} = +\,num.\log(-1.8680093 + 0.006864937\,t)$. for $t = 0^0$ to 100^0

$+ c\,\beta^{\tau} = -\,num.\log(0.0207601 - 0.005950708\,t)$ . . for $t = 100^0$ to 200^0

*) Reichsgesetzblatt 1871, P. 122.

**) log = Briggs's or common logarithm.

6) *The vacuum*, which is formed by the condensation of steam, can either be expressed in actual atmospheres or in kilos per □cm i. e. in new atmospheres. In the first case a vacuum of Vacuum.

$$1 \ old \ atm. = 76 \ cm$$

in the second case of

$$1 \ kilo \ per \ \square cm = 1 \ atm. = 73.551 \ centimetres$$

of the scale of a mercurial barometer.

7) *The pressure* prevailing in the approximate vacuum of condensers and the parts of cylinders in connection with them is called *the backpressure a.* It is always given in kilos per □cm. Back pressure a.

8) The weight in kilos of one cubic metre of saturated steam is called its *density* γ. According to ZEUNER*) it can be almost exactly found by means of the empirical formula Density of steam.

$$\gamma = a p^{\frac{1}{n}} \ kilos \ldots \ldots \ldots \ldots \ldots (24)$$

For the pressure p in old atm.

$$\alpha = 0,6061; \ \frac{1}{n} = 0,9393$$

and for the pressure p in kilos per □cm. PINZGER gives

$$\alpha = 0,5877; \ \frac{1}{n} = 0,939$$

9) *The volume* in cubic metres taken up by 1 kilo of saturated steam is called the *specific volume s*. It is of course the reciprocal of the density Specific Volume of steam.

$$s = \frac{1}{\gamma}; \ \gamma = \frac{1}{s}$$

10) *The volume* in cbmetres taken up by 1 kilo of moist steam or 1 kilo of a mixture of steam and water is called *the specific volume v* of the mixture. Specific volume of mixture v.

11) The volume of 1 kilo of water = 0.001 cbmetres at + 4° and is called the *specific volume of water w*. As water only expands 4.2 % or about $^1/_{25}$ of its volume when heated from its (practically) mean temperature of 15° to 100°, according to KOPP**), and can only be compressed from atmosphere to atmosphere of pressure 0.00005 of its volume at 0° and 0.000044 of its volume at 53° according to GRASSI***), the specific volume of water as compared to that of the steam evolved from it is regarded by engineers as *constant*, $w = 0.001$ cbmetres. Specific volume of water.

*) G. ZEUNER. Mechanische Wärmetheorie. Leipzig 1877. P. 294.

**) Muspratts theoretische, practische und analytische Chemie, fortgesetzt von B. KERL. II. Edition. Vol. 5. P. 861. Brunswick 1870.

***) Annales de chimie et de physique. III. Series. Vol. 31.

§ 10.

Generation of saturated steam.

Total heat λ.

1) To convert 1 kilo of water at 0^0 into saturated steam of t^0, a *total heat* of λ units of heat must be communicated to it, of which one part

q heats the water to t^0, another part, *the internal latent heat* ϱ changes its state of aggregation, and finally a third part, *the external latent heat* $A p u$ enables the steam to make room for itself.

Determination of the total heat.

2) REGNAULT*) bases the following empirical formula for the *determination of the total heat* upon experiments.

$$\lambda = 606.5 + 0.305\, t \quad \ldots \ldots \ldots \ldots \quad (25)$$

for $t = 100^0$

$$\lambda = 606.5 + 30.5 = 637 \text{ units of heat.}$$

Heat of the liquid q (called by Rankine h).

3) *The heat of the liquid* q, which heats the water to the point at which the generation of steam can begin, that is, produces the sensible energy is, likewise according to REGNAULT

$$q = t + 0.00002\, t^2 + 0.0000003\, t^3 \quad \ldots \ldots \ldots \quad (26)$$

for $t = 100^0$:

$$q = 100 + 0.2 + 0.3 = 100.5 \text{ units of heat.}$$

Internal latent heat ϱ.

4) *The internal latent heat* ϱ, which produces the work of translation by so far loosening the cohesion of the fluid particles that they pass beyond the range of their mutual attraction (see § 4, 10), is according to ZEUNER**)

$$\varrho = 575.4 - 0.791\, t \quad \ldots \ldots \ldots \ldots \quad (27)$$

for $t = 100^0$:

$$\varrho = 575.4 - 79.1 = 496.3 \text{ units of heat.}$$

External latent heat.

5) *The external latent heat* $A p u$ must produce the external work necessary to overcome the pressure p on the surface of the water which is equal to the pressure of the steam being formed. If we call $u = s - w$ the difference in cubic metres between s the volumes of a kilo of the steam and w that of a kilo of water, then this external work amounts in kilogrammetres to

$$p u = A p u \text{ units of heat}$$

and according to ZEUNER

$$A p u = 31.1 + 1.096\, t - q \quad \ldots \ldots \ldots \quad (28)$$

for $t = 100^0$:

$$A p u = 31.1 + 109.6 - 100.5 = 40.2 \text{ units of heat.}$$

*) Relation des expériences entreprises pour déterminer les lois et les données physiques nécessaires au calcul des machines à feu. Vol. I u. II. Paris 1862.

**) G. ZEUNER. Grundzüge der mechanischen Wärmetheorie. P. 283. Leipzig 1877.

Combination of the above.

6) From the preceding we get

$$\lambda = q + \varrho + A\,p\,u \quad \dots\dots\dots\dots \quad (29)$$

for $t = 100^0$:

$$\lambda = 100.5 + 496.3 + 40.2 = 637 \text{ units of heat.}$$

Heat of the steam J.

7) As the quantity of heat $A\,p\,u$ is immediately converted into work, it follows that only the "heat of the liquid" q and the internal latent heat ϱ are contained in the steam when formed. These two parts q and ϱ are therefore generally combined under the term *internal heat* or *"heat of the steam"* I

$$I = q + \varrho = \lambda - A\,p\,u \quad \dots\dots\dots\dots \quad (30)$$

for $t = 100^0$

$$I = 100.5 + 496.3 = 637 - 40.2 = 596.8 \text{ units of heat.}$$

Heat of Evaporation r.

8) Only ϱ and $A\,p\,u$ have to do with the evaporation of the water, because q does no more than raise its temperature. To ϱ and $A\,p\,u$ together the name *heat of evaporation* r has therefore been given

$$r = \varrho + A\,p\,u = \lambda - q \quad \dots\dots\dots\dots \quad (31)$$

for $t = 100^0$

$$r = 496.3 + 40.2 = 637 - 100.5 = 536.5 \text{ units of heat.}$$

Abbreviated formula for r.

9) If we put

$$r = \lambda - q$$

we may write, by Eq. 25 and 26

$$r = 606.5 + 305\,t - (t + 0.00002\,t^2 + 0.0000003\,t^3)$$

$$r = 606.5 - 0.695\,t - 0.00002\,t^2 - 0.0000003\,t^3$$

for which CLAUSIUS*) gives the much shorter and almost equally exact formula

$$r = 607 - 0.708\,t \quad \dots\dots\dots\dots \quad (32)$$

According to this formula r for $t = 100^0$ is

$$r = 607 - 70.8 = 536.2 \text{ units of heat,}$$

so that the difference between the result we arrive at by this expression and that of REGNAULT's much longer one is only 0.3 units of heat for $t = 100^0$.

Explanatory scheme of the quantities of heat.

10) The following scheme illustrates the distribution of the different quantities of heat to be taken into account when evaporation is considered; the values given are those corresponding to $t = 100^0$ (see § 4, 12).

Total heat λ (637 units of heat)

heat of the liquid q (100.5 units of heat) Internal latent heat ϱ (496.3 units) External latent heat $A\,p\,u$ (40.2 units)

heat of the steam I (596.8 units   Heat of evaporation r (536.5 units)

Notes on Table of properties of steam.

11) These values, for the most important steam pressures, are given in the annexed table, in which the numbers for pressures

) R. CLAUSIUS. Die mechanische Wärmetheorie. P. 137 u. 281. Brunswick 1876.

Table of the properties of saturated steam.

Pressure in kilos per □cm p	Pressure in mm. of Mercury	Temperature centigrade t	heat of the liquid q	internal latent heat ϱ	external latent heat Apu	heat of the steam J	heat of evaporation r	total heat λ	volume in cb. m. of a kilo of steam s	Weight in kilos of a cb. m. of steam γ
1	2	3	4	5	6	7	8	9	10	11
0,006	4,60	0,00	0,00	575,43	31,07	575,43	606,50	606,50	210,652	0,0047
0,007	4,94	1,00	1,00	574,65	31,15	575,65	605,80	606,80	196,672	0,0051
0,007	5,30	2,00	2,00	573,88	31,23	575,88	605,11	607,11	183,699	0,0054
0,008	5,69	3,00	3,00	573,10	31,31	576,10	604,41	607,41	171,718	0,0058
0,008	6,10	4,00	4,00	572,32	31,39	576,32	603,72	607,72	160,593	0,0062
0,009	6,53	5,00	5,00	571,55	31,48	576,55	603,02	608,02	150,224	0,0067
0,010	7,00	6,00	6,00	570,77	31,56	576,77	602,33	608,33	140,637	0,0071
0,010	7,49	7,00	7,00	569,99	31,64	576,99	601,63	608,63	131,657	0,0076
0,011	8,02	8,00	8,00	569,21	31,72	577,21	600,94	608,94	123,408	0,0081
0,012	8,57	9,00	9,00	568,43	31,81	577,43	600,24	609,24	115,666	0,0086
0,012	9,16	10,00	10,00	567,66	31,89	577,66	599,55	609,55	108,526	0,0092
0,013	9,79	11,00	11,00	566,88	31,98	577,88	598,85	609,85	101,863	0,0098
0,014	10,46	12,00	12,00	566,10	32,06	578,10	598,16	610,16	95,598	0,0105
0,015	11,16	13,00	13,00	565,31	32,15	578,31	597,46	610,46	89,790	0,0111
0,016	11,91	14,00	14,00	564,53	32,23	578,53	596,76	610,77	84,411	0,0118
0,017	12,70	15,00	15,00	563,75	32,32	578,75	596,07	611,07	79,346	0,0126
0,018	13,54	16,00	16,01	562,97	32,40	578,98	595,37	611,38	74,673	0,0134
0,020	14,42	17,00	17,01	562,19	32,49	579,20	594,68	611,68	70,254	0,0142
0,021	15,36	18,00	18,01	561,40	32,58	579,41	593,98	611,99	66,158	0,0151
0,022	16,35	19,00	19,01	560,62	32,67	579,63	593,29	612,29	62,336	0,0160
0,024	17,39	20,00	20,01	559,83	32,75	579,84	592,59	612,60	58,724	0,0170
0,025	18,49	21,00	21,01	559,05	32,84	580,06	591,89	612,90	55,370	0,0181
0,027	19,66	22,00	22,01	558,26	32,93	580,27	591,20	613,21	52,239	0,0191
0,028	20,89	23,00	23,01	557,48	33,02	580,49	590,50	613,51	49,301	0,0203
0,030	22,18	24,00	24,02	556,69	33,11	580,71	589,80	613,82	46,550	0,0215
0,032	23,55	25,00	25,02	555,91	33,20	580,93	589,11	614,12	43,965	0,0227
0,034	24,99	26,00	26,02	555,12	33,29	581,14	588,41	614,43	41,553	0,0241
0,036	26,50	27,00	27,02	554,33	33,38	581,35	587,71	614,73	39,275	0,0255
0,038	28,10	28,00	28,02	553,54	33,47	581,56	587,02	615,04	37,146	0,0269
0,040	29,78	29,00	29,02	552,76	33,56	581,78	586,32	615,34	35,149	0,0284
0,043	31,55	30,00	30,03	551,97	33,66	582,00	585,62	615,65	33,272	0,0300
0,045	33,41	31,00	31,03	551,18	33,75	582,21	584,93	615,95	31,505	0,0317
0,048	35,36	32,00	32,03	550,39	33,84	582,42	584,23	616,26	29,849	0,0335
0,051	37,41	33,00	33,03	549,60	33,93	582,63	583,53	616,56	28,289	0,0353
0,054	39,56	34,00	34,03	548,81	34,02	582,84	582,83	616,87	26,821	0,0373
0,057	41,83	35,00	35,04	548,02	34,12	583,06	582,14	617,17	25,439	0,0393
0,060	44,20	36,00	36,04	547,23	34,21	583,27	581,44	617,48	24,137	0,0414
0,063	46,69	37,00	37,04	546,44	34,30	583,48	580,74	617,78	22,914	0,0436
0,067	49,30	38,00	38,04	545,64	34,40	583,68	580,04	618,09	21,761	0,0459
0,071	52,04	39,00	39,05	544,85	34,49	583,90	579,35	618,39	20,673	0,0484
0,075	54,91	40,00	40,05	544,06	34,59	584,10	578,65	618,70	19,646	0,0509
0,079	57,91	41,00	41,05	543,27	34,68	584,32	577,95	619,00	18,679	0,0535
0,083	61,05	42,00	42,06	542,47	34,78	584,53	577,25	619,31	17,764	0,0563
0,087	64,35	43,00	43,06	541,68	34,87	584,74	576,55	619,61	16,903	0,0592
0,092	67,79	44,00	44,06	540,88	34,97	584,94	575,86	619,92	16,087	0,0622
0,097	71,39	45,00	45,07	540,09	35,06	585,16	575,16	620,22	15,318	0,0653
0,102	75,16	46,00	46,07	539,30	35,16	585,37	574,46	620,53	14,590	0,0685
0,108	79,10	47,00	47,07	538,50	35,26	585,57	573,76	620,83	13,903	0,0719
0,113	83,20	48,00	48,08	537,71	35,35	585,78	573,06	621,14	13,252	0,0755
0,119	87,50	49,00	49,08	536,91	35,45	585,99	572,36	621,44	12,635	0,0791
0,125	91,98	50,00	50,09	536,12	35,54	586,21	571,66	621,75	12,052	0,0830
0,131	96,66	51,00	51,09	535,32	35,64	586,41	570,96	622,05	11,499	0,0870
0,138	101,54	52,00	52,10	534,53	35,74	586,63	570,26	622,36	10,977	0,0911
0,145	106,64	53,00	53,10	533,73	35,83	586,83	569,56	622,66	10,480	0,0954
0,152	111,95	54,00	54,11	532,93	35,93	587,04	568,86	622,97	10,009	0,0999

Pressure in kilos per □cm p	Pressure in mm of Mercury	Temperature centigrade t	heat of the liquid q	internal latent heat ρ	external latent heat Apu	heat of the steam J	heat of evaporation r	total heat λ	volume in cb. m. of a kilo of steam s	Weight in kilos of a cb. m. of steam γ
1	2	3	4	5	6	7	8	9	10	11
0,160	117,48	55,00	55,11	532,14	36,03	587,25	568,16	623,27	9,5643	0,1046
0,168	123,24	56,00	56,11	531,34	36,12	587,45	567,46	623,58	9,1420	0,1094
0,176	129,25	57,00	57,12	530,54	36,22	587,66	566,76	623,88	8,7404	0,1144
0,184	135,51	58,00	58,13	529,75	36,31	587,88	566,06	624,19	8,3588	0,1196
0,193	142,02	59,00	59,13	528,95	36,41	588,08	565,36	624,49	7,9973	0,1250
0,202	148,79	60,00	60,14	528,15	36,51	588,29	564,66	624,80	7,6535	0,1307
0,212	155,84	61,00	61,14	527,35	36,61	588,49	563,96	625,10	7,3271	0,1365
0,222	163,17	62,00	62,15	526,55	36,71	588,70	563,26	625,41	7,0164	0,1425
0,232	170,79	63,00	63,15	525,76	36,80	588,91	562,56	625,71	6,7212	0,1488
0,243	178,71	64,00	64,16	524,96	36,90	589,12	561,86	626,02	6,4403	0,1553
0,254	186,95	65,00	65,17	524,16	37,00	589,33	561,16	626,32	6,1725	0,1620
0,266	195,50	66,00	66,17	523,36	37,09	589,53	560,46	626,63	5,9182	0,1690
0,278	204,38	67,00	67,18	522,56	37,19	589,74	559,75	626,93	5,6756	0,1762
0,290	213,60	68,00	68,19	521,77	37,29	589,96	559,05	627,24	5,4448	0,1837
0,303	223,17	69,00	69,19	520,97	37,38	590,16	558,35	627,54	5,2248	0,1914
0,317	233,09	70,00	70,20	520,17	37,48	590,37	557,65	627,85	5,0153	0,1994
0,331	243,39	71,00	71,21	519,37	37,57	590,58	556,95	628,15	4,8154	0,2077
0,345	254,07	72,00	72,22	518,57	37,67	590,79	556,24	628,46	4,6247	0,2162
0,361	265,15	73,00	73,22	517,78	37,76	591,00	555,54	628,76	4,4427	0,2251
0,376	276,62	74,00	74,23	516,98	37,86	591,21	554,84	629,07	4,2693	0,2342
0,392	288,52	75,00	75,24	516,18	37,95	591,42	554,14	629,37	4,1035	0,2437
0,409	300,84	76,00	76,25	515,38	38,05	591,63	553,43	629,68	3,9454	0,2535
0,426	313,60	77,00	77,25	514,58	38,14	591,83	552,73	629,98	3,7943	0,2635
0,444	326,81	78,00	78,26	513,79	38,24	592,05	552,03	630,29	3,6499	0,2740
0,463	340,49	79,00	79,27	512,99	38,33	592,26	551,32	630,59	3,5118	0,2847
0,482	354,64	80,00	80,28	512,19	38,42	592,47	550,62	630,90	3,3799	0,2959
0,502	369,29	81,00	81,29	511,40	38,52	592,69	549,91	631,20	3,2537	0,3073
0,523	384,44	82,00	82,30	510,60	38,61	592,90	549,21	631,51	3,1331	0,3192
0,544	400,10	83,00	83,31	509,80	38,70	593,11	548,51	631,81	3,0177	0,3314
0,566	416,30	84,00	84,32	509,01	38,79	593,33	547,80	632,12	2,9071	0,3440
0,589	433,04	85,00	85,33	508,21	38,88	593,54	547,10	632,42	2,8013	0,3570
0,612	450,34	86,00	86,34	507,42	38,97	593,76	546,39	632,73	2,7000	0,3704
0,637	468,22	87,00	87,35	506,62	39,06	593,97	545,69	633,03	2,6029	0,3842
0,662	486,69	88,00	88,36	505,83	39,15	594,19	544,98	633,34	2,5099	0,3984
0,688	505,76	89,00	89,37	505,03	39,24	594,40	544,27	633,64	2,4208	0,4131
0,714	525,45	90,00	90,38	504,24	39,33	594,62	543,57	633,95	2,3354	0,4282
0,742	545,78	91,00	91,39	503,44	39,42	594,83	542,86	634,25	2,2534	0,4438
0,771	566,76	92,00	92,40	502,65	39,51	595,05	542,16	634,56	2,1748	0,4598
0,800	588,41	93,00	93,41	501,86	39,59	595,27	541,45	634,86	2,0994	0,4763
0,830	610,74	94,00	94,43	501,07	39,68	595,50	540,74	635,17	2,0270	0,4933
0,862	633,78	95,00	95,44	500,27	39,76	595,71	540,04	635,47	1,9575	0,5109
0,894	657,54	96,00	96,45	499,48	39,85	595,93	539,33	635,78	1,8909	0,5288
0,927	682,03	97,00	97,46	498,69	39,93	596,15	528,62	636,08	1,8270	0,5473
0,962	707,26	98,00	98,47	497,89	40,02	596,36	537,92	636,39	1,7658	0,5663
1,000	735,51	99,09	99,58	497,05	40,10	596,63	537,15	636,72	1,7012	0,5878
1,033	760,00	100,00	100,50	496,29	40,20	596,79	536,50	637,00	1,6508	0,6058
1,250	919,39	105,41	105,98	492,00	40,66	597,98	532,67	638,65	1,3804	0,7244
1,500	1103,27	110,76	111,42	487,76	41,11	599,18	528,87	640,28	1,1631	0,8598
1,750	1287,14	115,42	116,15	484,06	41,49	600,21	525,55	641,70	1,0063	0,9937
2,000	1471,02	119,57	120,37	480,78	41,82	601,15	522,60	642,97	0,8877	1,1265
2,250	1654,90	123,31	124,18	477,81	42,12	601,99	519,93	644,11	0,7947	1,2583
2,500	1838,78	126,73	127,66	475,11	42,38	602,77	517,49	645,15	0,7198	1,3893
2,750	2022,65	129,87	130,87	472,62	42,62	603,49	515,23	646,11	0,6582	1,5193
3,000	2206,53	132,80	133,85	470,30	42,85	604,15	513,15	647,00	0,6066	1,6485
3,250	2390,41	135,53	136,64	468,14	43,05	604,78	511,19	647,84	0,5626	1,7775

Pressure in kilos per □cm p	Pressure in mm. of Mercury	Temperature centigrade t	heat of the liquid q	internal latent heat ρ	external latent heat Apu	heat of the steam J	heat of evaporation r	total heat λ	volume in cb. m. of a kilo of steam s	Weight in kilos of a cb. m. of steam γ
1	2	3	4	5	6	7	8	9	10	11
3,50	2574,29	138,10	139,27	466,11	43,24	605,38	509,35	648,62	0,5248	1,905
3,75	2758,16	140,52	141,75	464,19	43,41	605,94	507,61	649,36	0,4919	2,033
4,00	2942,04	142,82	144,10	462,38	43,58	606,48	505,96	650,06	0,4630	2,160
4,25	3125,92	145,00	146,34	460,65	43,74	606,99	504,39	650,73	0,4373	2,287
4,50	3309,80	147,09	148,47	459,00	43,88	607,48	502,89	651,36	0,4145	2,412
4,75	3493,67	149,08	150,52	457,43	44,02	607,95	501,45	651,97	0,3940	2,538
5,00	3677,55	150,99	152,48	455,92	44,15	608,40	500,07	652,55	0,3754	2,664
5,25	3861,43	152,83	154,36	454,47	44,28	608,83	498,75	653,11	0,3586	2,789
5,50	4045,31	154,59	156,18	453,07	44,40	609,25	497,47	653,65	0,3433	2,913
5,75	4229,18	156,30	157,93	451,72	44,51	609,65	496,24	654,17	0,3292	3,038
6,00	4413,06	157,94	159,62	450,42	44,62	610,04	495,05	654,67	0,3164	3,161
6,25	4596,94	159,54	161,26	449,16	44,73	610,42	493,89	655,16	0,3045	3,284
6,50	4780,82	161,08	162,85	447,94	44,83	610,79	492,78	655,63	0,2934	3,408
6,75	4964,69	162,57	164,39	446,76	44,93	611,15	491,69	656,08	0,2832	3,531
7,00	5148,57	164,03	165,89	445,61	45,02	611,50	490,64	656,53	0,2737	3,654
7,25	5332,45	165,44	167,35	444,50	45,11	611,85	489,61	656,96	0,2648	3,776
7,50	5516,33	166,81	168,76	443,41	45,20	612,17	488,61	657,38	0,2565	3,899
7,75	5700,20	168,15	170,15	442,35	45,29	612,50	487,64	657,79	0,2488	4,019
8,00	5884,08	169,46	171,49	441,32	45,37	612,81	486,69	658,18	0,2415	4,141
8,25	6067,96	170,73	172,81	440,32	45,45	613,13	485,76	658,57	0,2346	4,263
8,50	6251,84	171,98	174,09	439,33	45,53	613,42	484,86	658,95	0.2281	4,384
8,75	6435,71	173,19	175,35	438,37	45,60	613,72	483,97	659,32	0,2220	4,504
9,00	6619,59	174,38	176,58	437,43	45,67	614,01	483,11	659,69	0,2162	4,625
9,25	6803,47	175,54	177,78	436,51	45,74	614,29	482,26	660,04	0,2107	4,746
9,50	6987,35	176,68	178,96	435,62	45,81	614,58	481,43	660,39	0,2055	4,866
9,75	7171,22	177,79	180,11	434,73	45,88	614,84	480,62	660,73	0,2005	4,987
10,00	7355,10	178,89	181,24	433,87	45,95	615,11	479,82	661,06	0,1958	5,107
10,25	7538,98	179,96	182,35	433,02	46,01	615,37	479,03	661,39	0.1913	5,227
10,50	7722,86	181,01	183,44	432,19	46,07	615,63	478,26	661,71	0,1870	5,348
10.75	7906,73	182,04	184,51	431,38	46,13	615,89	477,51	662,62	0,1830	5,464
11,00	8090,61	183,05	185,56	430,58	46,19	616,14	476,77	662,33	0,1791	5,583
11,25	8274,49	184,05	186,60	429,79	46,25	616,39	476,04	662,63	0,1753	5,704
11,50	8458,37	185,03	187,61	429.01	46,31	616,62	475,32	662,93	0,1717	5,824
11.75	8642,24	185,99	188,61	428,25	46,36	616,86	474,62	663,23	0,1683	5,942
12,00	8826,12	186,93	189,59	427,51	46,41	617,10	473,92	663,51	0,1650	6,061
12,25	9010,00	187,87	190,56	426,77	46,47	617,27	473,24	663,80	0,1618	6,180
12,50	9193,88	188,78	191,51	426,05	46,52	617,56	472,57	664,08	0,1588	6,297
12,75	9377,75	189,68	192,45	425,33	46,57	617,78	471,90	664,35	0,1559	6,414
13,00	9561,63	190,57	193,38	424,63	46,62	618,01	471,25	664,62	0,1531	6,532
13,25	9745,51	191,45	194,29	423,94	46,67	618,23	470,60	664,89	0,1503	6.653
13,50	9929,39	192,31	195,18	423,25	46,72	618,43	469,97	665,15	0,1477	6,770
13,75	10113,26	193,16	196,07	422,58	46,76	618,65	469,34	665,41	0,1452	6,887
14,00	10297,14	194,00	196,94	421,92	46,81	618,86	468,73	665,67	0,1428	7,003
14,25	10481,02	194,83	197,81	421,26	46,85	619,07	468,12	665,92	0,1404	7,122
14,50	10664,90	195,64	198,66	420,61	46,90	619,27	467,51	666,17	0,1381	7,241
14,75	10848,77	196,45	199,49	419,98	46,94	619,47	466,92	666,42	0,1359	7,358
15,00	11032,65	197,24	200,32	419,35	46,99	619,67	466,33	666,66	0,1338	7,474
16,00	11768,13	200,31	203,54	416,91	47,14	620,45	464,05	667,59	0,1295	7,724
17,00	12503,67	203,24	206,60	414,59	47,30	621,19	461,89	668,49	0,1223	8,176
18,00	13239,18	206,05	209,52	412,39	47,43	621,91	459,82	669,34	0,1159	8,627
19,00	13974,69	208,75	212,32	410,28	47,57	622,60	457,85	670,17	0,1102	9,076
20,00	14710,20	211,34	215,06	408,23	47,67	623,29	455,90	670,96	0,1049	9,527
21,00	15445,71	213,83	217,68	406,26	47,78	623.94	454,04	671,72	0,1003	9,975
22,00	16181,22	216,23	220,20	404,36	47,89	624,56	452,25	672,45	0,0960	10,416
23,00	16916,73	218,55	222,64	402,53	47,99	625,17	450,52	673,16	0,0921	10,860

Pressure in kilos per □cm p	Pressure in mm. of Mercury	Temperature centigrade t	heat of the liquid q	internal latent heat ϱ	external latent heat Apu	heat of the steam J	heat of evaporation r	total heat λ	volume in cb. m. of a kilo of steam s	Weight in kilos of a cb. m. of steam γ
1	2	3	4	5	6	7	8	9	10	11
24,00	17652,24	220,79	224,99	400,76	48,09	625,75	448,85	673,84	0,0885	11,313
25,00	18387,75	222,96	227,28	399,04	48,18	626,32	447,22	674,50	0,0854	11,745
26,00	19123,26	225,07	229,50	397,37	48,28	626,87	445,65	675,15	0,0821	12,185
27,00	19858,77	227,12	231,67	395,75	48,35	627,42	444,10	675,77	0,0792	12,625
28,00	20594,28	229,11	233,77	394,17	48,44	627,94	442,61	676,38	0,0766	13,063
29,00	21329,79	231,05	235,82	392,64	48,51	628,46	441,15	676,97	0,0741	13,501
30,00	22065,30	232,95	237,83	391,15	48,57	628,98	439,72	677,55	0,0718	13,938

between 0 and 1 kilo. per □cm were calculated by PINZGER*) according to ZEUNER's theories and with the help of REGNAULT's original tables. The numbers from 1 to 15 kilos per □cm (except those of s and γ) were taken from FLIEGNER's table**). These last values (s and γ) have been re-calculated by PINZGER for $A = \frac{1}{424}$, because the value of $A = \frac{1}{436}$ deduced by REGNAULT from the velocity of sound and adopted by FLIEGNER has been in the meantime shewn to be incorrect. The numbers from 15 to 30 kilos per □cm were determined by ALTHANS***) for an enlargement of FLIEGNER's table, so that in these the values of s and γ are based upon $A = \frac{1}{436}$. It was not considered worth while to re-calculate these for $A = \frac{1}{424}$, because they are to a certain extent doubtful as it is, REGNAULT's experiments having only embraced temperatures up to 200°. They are included here in default of better, exacter, and equally for-reaching data.

§ 11.

Principal laws of the changes of state of mixtures of steam and water.

1) In considering the changes of state oi saturated steam, we must always assume a mixture of steam and water, if the steam **Mixture of steam and water.**

*) MEHRTENS. Technische Mechanik. IX. Mechanik der gas- und dampfförmigen Körper, bearbeitet von L. PINZGER. P. 826. Berlin 1887.

**) Civilingenieur, Vol. XX, P. 442. 1874.

***) Zeitschrift für Berg-, Hütten- und Salinenwesen, Vol. XXIII, P. 280. 1875.

is to remain in a state of saturation at every moment. Either moist steam alone, or the contents of a boiler consisting of water together with dry or moist steam, is to be regarded as such a mixture of steam and water.

Volume of mixture v.

2) **I. General Equations for reversible changes of state of a mixture of steam and water.** *The volume v of 1 kilo of such a mixture,* whose pressure is p can be easily calculated, when the amount of water it contains (or its "dryness-fraction") is known. If x is the weight of the dry steam and therefore $1-x$ the weight of the water, while s is the specific volume of the steam and w that of the water, the volume of the mixture must be

$$v = xs + (1-x)w = x(s-w) + w$$

or, as according to § 10, 5: $s - w = u$

$$v = xu + w \text{ cubic metres} \quad \ldots \ldots \ldots \ldots \quad (33)$$

The values of u are found from the Table on p. 28 to 31 by deducting 0.001 from the values there given for s. For instance, the volume of 1 kilo of steam containing 5 % of moisture ($x = 0.95$) for $p = 8$ kilos per □cm

$$v = 0.95\,(2.415 - 0.001) + 0.001$$

$$v = 0.95 \cdot 2.414 + 0.001 = 2.2943 \text{ cbm.}$$

Internal work dU of the mixture.

3) *The internal work dU* produced by *an infinitely small reversible change of state of a mixture of steam and water* is computed by determining the heat value AU of the whole internal work contained in the mixture. To do this we must add the "heat of the liquid" of the $1-x$ kilos of water, i. e.

$$(1-x)\,q \quad \text{units of heat}$$

to the "heat of the steam" (I) of the x kilos of steam,

$$xI \quad \text{units of heat}$$

$$AU = (1-x)\,q + xI$$

$$AU = q + x\,(I - q)$$

or, as by Eq. 30 $\quad I - q = \varrho$

$$AU = q + x\varrho \quad \text{units of heat} \quad \ldots \ldots \quad (34)$$

We therefore have, for the heat value of the internal work of an infinitely small reversible change of state

$$A\,dU = dq + d\,(x\varrho) \quad \ldots \ldots \ldots \ldots \quad (35)$$

and for the internal work itself

$$dU = \frac{1}{A}\Big[dq + d\,(x\varrho)\Big] \quad \ldots \ldots \ldots \ldots \quad (35^a)$$

4) *The quantity of heat* dQ to be communicated *to 1 kilo of a mixture of steam and water for an infinitely small reversible change of state* is given by the general equation **Derivation of different values for the heat quantity dQ.**

$$dQ = A(dW + dJ + dL)$$

by Eq. 2[a]

$$dW + dJ = dU$$

and by Eq. 35

$$A\,dU = dq + d(x\varrho)$$

Further, — as during a reversible change of state the pressure p must always equal the external pressure, therefore the external work dL, corresponding to the expansion dv (the volume of water in the mixture being regarded as constant) is

$$dL = p\,dv$$

whence follows

$$dQ = dq + d(x\varrho) + Ap\,dv \quad \ldots\ldots\ldots\ldots \quad (36)$$

If in this equation we express the weight x in terms of the volume v by Eq. 33,

$$x = \frac{v - w}{u}$$

$$dQ = dq + d\left[\frac{\varrho}{u}(v - w)\right] + Ap\,dv$$

and differentiate, taking q and $\frac{\varrho}{u}$ as functions of p, we get

$$dQ = \frac{d}{dp}\left[q - \frac{\varrho}{u}w + \frac{\varrho}{u}v\right]dp + \left(\frac{\varrho}{u} + Ap\right)dv;$$

or, substituting in the second term by Eq. 31,

$$\varrho = r - Apu$$

$$dQ = \frac{d}{dp}\left[q - \frac{\varrho}{u}w + \frac{\varrho}{u}v\right]dp + \frac{r}{u}\,dv \quad \ldots\ldots \quad (36^{a})$$

If now we express in Eq. 36, the volume v in terms of the weight x, and differentiate Eq. 33 in which w is constant

$$dv = d(xu)$$

we may put $\qquad Ap\,dv = Ap\,d(xu)$

or $\qquad Ap\,dv = A\,d(pux) - A\,xu\,dp$

hereupon Eq. 36 takes the following form

$$dQ = dq + d(x\varrho) + A\,d(pux) - A\,xu\,dp.$$

Substituting by Eq. 31

$$r = \varrho + Apu,$$

the second and third terms of this equation may be combined as $d(xr)$. For the last term we may substitute after some simplifications

$$A\,xu\,dp = \frac{xr}{T}\,dt$$

consequently

$$dQ = dq + d(xr) - \frac{xr}{T}\,dt = dq + T\,d\left(\frac{xr}{T}\right) \quad \ldots \quad (36^b)$$

Calling the specific heat of water C and regarding this as constant, which it is *not* by § 5, 6, we may put, in the above equation

$$dq = Cdt\,.$$

Differentiating the second term, as indicated, we get

$$dQ = Cdt + r\,dx + x\,dr - x\frac{r}{T}dt$$

or adding and subtracting $x\,Cdt$ on the right hand and arranging the terms accordingly,

$$dQ = (1-x)\,Cdt + rdx + x\left(C + \frac{dr}{dt} - \frac{r}{T}\right)dt$$

and putting

$$C + \frac{dr}{dt} - \frac{r}{T} = h$$

we have

$$dQ = (1-x)\,Cdt + r\,dx + x\,h\,dt \quad \ldots \ldots \ldots \quad (36^c)$$

In this expression

$(1-x)\,Cdt$ is the heat quantity which raises the temperature of the $1-x$ kilos of fluid dt degrees,

$r\,dx$ the heat quantity necessary for evaporating dx kilos of the fluid, and

$x\,h\,dt$ the heat quantity which raises the temperature of the x kilos of steam dt degrees and produces the increment of volume dv corresponding to this rise of temperature.

Specific heat h of saturated steam.

5) From the foregoing it is obvious that h represents *the specific heat of saturated steam* under the particular circumstances of a change of volume and pressure (caused by communication of heat) such that the steam remains *just* saturated. ZEUNER*) calls this "*the specific heat of saturated steam under constant steam quantity*". CLAUSIUS**) gives the following formula for determining h at different temperatures t, in which he takes the specific heat of water at $t = 100^0$ as $C = 1.013$***), (whereas ZEUNER****), taking into consideration that engineers always have to do with higher temperatures than 100^0, puts $C = 1.0224$)

*) G. ZEUNER. Die mechanische Wärmetheorie. Leipzig 1877. P. 312.

**) R. CLAUSIUS. Die mechanische Wärmetheorie. Brunswick 1876. P. 136.

***) The same.

****) G. ZEUNER. Die mechanische Wärmetheorie. Leipzig 1877. P. 322.

$$h = 0.305 - \frac{r}{T} \quad \ldots\ldots\ldots\ldots\ldots \quad (37)$$

Substituting in this the value of r by Eq. 32 we have

$$h = 0.305 - \frac{607 - 0.708\,t}{273 + t}$$

and this equation may be brought into the following simpler form

$$h = 1.013 - \frac{800.3}{273 + t}$$

according to which CLAUSIUS calculated the values below.

$t = 0^0$	50^0	100^0	150^0	200^0
$h = -$ 1,916	$-$ 1,465	$-$ 1,133	$-$ 0,879	$-$ 0,676

The negative, and indeed large negative, values of the specific heat of saturated steam are explained by the circumstance that during the compression of the steam the work expended produces more heat than suffices to keep the steam at the saturation temperature corresponding to its new density. If it is only to be heated so far that it remains just saturated, part of the heat generated (by the work of compression) must be withdrawn from it. Consequently, during expansion more heat is converted into work than is required to keep the steam just cooled to its saturation point. If it is to remain at this point, heat must be imparted to it during expansion. If the changes of state of saturated steam take place in a nonconducting vessel, it becomes superheated by compression and is partly condensed during expansion.

6) *The external work or work of expansion* dL performed by 1 kilo of a mixture of steam and water during an infinitely small reversible change of state, dQ being the communicated heat quantity, is found from Eq. 36: **External work dL of the mixture.**

$$Ap\,dv = d\,Q - \Big[dq + d\,(x\varrho)\Big] = A\,d\,L\,;$$

substituting the value of $d\,Q$ from Eq. 36^b we get

$$Ap\,dv = dq + T\,d\left(\frac{x\,r}{T}\right) - \Big[dq + d\,(x\,\varrho)\Big] = T\,d\left(\frac{x\,r}{T}\right) - d\,(x\,\varrho) = A\,d\,L$$

hence

$$d\,L = \frac{1}{A}\left[T\,d\left(\frac{x\,r}{T}\right) - d\,(x\,\varrho)\right] \quad \ldots\ldots\ldots \quad (38)$$

Isothermal for mixed steam.

7) **II. Changes of state with temperature constant.** The curve illustrating the variations of the pressure with the volume of a gas, at constant temperature t, is called the Isothermal, § 6, 16. As, by § 9, 5 the pressure p of mixtures of steam and liquid depends upon the temperature alone, therefore with constant temperature the pressure must also be constant. Hence it follows that *the isothermal for mixtures of steam and water is a straight line parallel to the axis of abscissæ* (Plate 1, Fig. 4, Line I).

External work L.

8) *The external work L* performed by 1 kilo of a mixture of steam and water in expanding under constant temperature from v to v_1 while the dryness fraction changes from x to x_1 is determined from

$$d L = p\, dv$$

$$L = \int_v^{v_1} p\, dv = p\,(v_1 - v)$$

or, substituting

$$v_1 = x_1\, u + w \text{ and } v = x\, u + w$$

$$L = p\, u\,(x_1 - x) \text{ kilogrammetres} \quad \dots\dots\dots (39)$$

Heat quantity Q.

9) *The heat quantity Q* to be imparted to 1 kg of mixed steam in order that it may perform the external work expressed in Eq. 39, is composed of the heat value $A\,L$ of the latter plus the heat value of the internal work. This last is given as follows by Eq. 35, in which $d\,q$ now equals zero and ϱ is constant because the temperature is constant,

$$A\int_U^{U_1} d\,U = \int \Big[dq + d\,(x\,\varrho)\Big] = \varrho \int_x^{x_1} dx$$

$$A\,(U_1 - U) = \varrho\,(x_1 - x)$$

in which U_1 and x_1 belong to the terminal state. Consequently

$$Q = A\,L + A\,(U_1 - U)$$

$$Q = A\,p\,u\,(x_1 - x) + \varrho\,(x_1 - x) = (A p\, u + \varrho)\,(x_1 - x)$$

or, referring to Eq. 31:

$$Q = r\,(x_1 - x) \text{ units of heat} \quad \dots\dots\dots (40)$$

Heat quantity Q.

10) **III. Changes of State with constant volume.** From Eq. 33 it follows that when v is constant $x\,u$ is also constant, or that, if x and u represent the initial, and $x_1\ u_1$ the terminal state of a mixture of steam and water

$$x\,u = x_1\, u_1\,;\ x_1 = \frac{u}{u_1}\, x$$

The heat quantity Q which must be imparted to 1 kilo of mixed steam to produce any particular change of state with constant volume is simply the heat value of the internal work performed, because $dL = 0$. Then we have, by Eq. 35

$$dQ = A\,dU = dq + d(x\varrho)\,;$$

integrating this and referring q and ϱ to the initial, and $q_1\ \varrho_1$ to the terminal state,

$$Q = q_1 - q + x_1\varrho_1 - x\varrho\,.$$

Substituting in this the value of x_1, as determined above, we find

$$Q = q_1 - q + x u\left(\frac{\varrho_1}{u_1} - \frac{\varrho}{u}\right) \text{ units of heat} \ldots\ldots (41)$$

11) With the help of this equation the time can be calculated in which the pressure of steam in a perfectly closed and uniformly fired boiler will rise by a certain amount, as the following example shews. The pressure in a torpedo-boat's boiler stands at one atmosphere and is to be raised to 12 atmospheres. The boiler contains 3.5 cubic metres = 3500 kilos of water and 2 cubic metres = 2×0.5878 (see table on Page 29) = 1.1756 kilos of steam. The dryness fraction of the mixture is therefore $x = \dfrac{1.1756}{3500 + 1.1756} = 0.0003359$. To determine the heat-quantity Q which must be imparted to a kilo of this mixture to raise its pressure from 1 to 12 atmospheres with constant volume, we have only to substitute in Eq. 41 the respective numerical values from the Table on Page 28 and 31, thus

Zeuner's Boiler problem.

$$Q = 189.59 - 99.58 + 0.0003359 \times 1.7002\left(\frac{427.51}{0.1640} - \frac{497.05}{1.7002}\right) = 88.7 \text{ units of heat}$$

So that we must impart to the whole 3501.1756 kilos of mixture in the boiler 3501.1756×88.7 units of heat. If the boiler has 90 □metres of heating surface, consisting chiefly of brass tubes which will transmit at least 300 units of heat per □metre per minute from the furnace gases to the water, the required pressure will be reached in

$$\frac{3501.1756 \times 88.7}{90 \times 300} = 11.5 \text{ Minutes}$$

assuming the boiler to be perfectly closed.

12) **IV. Change of state with constant dryness fraction (x).** If x is constant, we can regard Eq. 33, in which u is a function of p, as the equation of a curve whose coordinates are p and v. Substituting in Eq. 33, the density of the steam from Eq. 24

Saturation Curve.

$$\gamma = a p^{\frac{1}{n}} = \frac{1}{s} \text{ so that } u = s - w = \frac{1}{\gamma} - w = \frac{1}{a p^{\frac{1}{n}}} - w,$$

we get, as the Equation of *the Saturation Curve*

$$v = (1 - x) w + \frac{x}{a p^{\frac{1}{n}}} \quad \ldots\ldots\ldots\ldots \quad (42)$$

This curve is a hyperbola (see Plate 1, Fig. 4, Curve IV) the one assymptote of which is the axis of x, the other assymptote a straight line drawn parallel to the axis of y at the distance $(1 - x)\, w$ from the origin of coordinates. For one kilo of dry steam, $x = 1$, and the above equation becomes

$$p v^n = a \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (42^a)$$

in which, according to PINZGER,*)
for p in kilos per □cm,

$n = 1.064963$ to 1.065 **)
$a = 1.76133$

for p in "old atmospheres" ZEUNER***) gives

$n = 1.0646$
$a = 1.704$

RANKINE****) puts, for p in ℔ per sq. inch and v in cubic ft. (english) at pressures up to 8 atmospheres and expanding up to 16 times in jacketed cylinders,

$$n = 1.0625 = \frac{17}{16}$$

$$a = 475$$

13) *The heat-quantity dQ, to be imparted to 1 kilo of mixed steam* for an infinitely small change of state with constant dryness-fraction, is by Eq. 36^c, as $dx = 0$

$$dQ = \Big[(1 - x) C + x h\Big] dt \quad \ldots\ldots\ldots\ldots \quad (43)$$

for dry steam $\quad x = 1$

$$dQ = h\, dt \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (43^a)$$

So long as $\Big[(1 - x) C + x h\Big] < 0$, i. e. so long as $x > \frac{C}{C - h}$, a negative value of dt corresponds to a positive value of dQ and vice versâ.

But by Eq. 37 $h = 0.305 - \frac{r}{T}$ is always negative because of

*) MEHRTENS. Technische Mechanik. Berlin 1887. P. 832.

**) GIZYCKI. Untersuchungen der Dampfmaschinen etc. in Düsseldorf. 1880. Aachen 1881. P. 9.

***) G. ZEUNER. Mechanische Wärmetheorie. Leipzig 1877. P. 294.

****) J. W. M. RANKINE. The steam engine. London 1873. P. 403.

the large value of $\frac{r}{T}$. Hence it follows that for $x = 1$ (that is for dry steam) with *decreasing* temperature and therefore *decreasing* pressure *dQ is positive* and with *increasing* temperature and pressure dQ becomes *negative.* This remains the case so long as

$$x > 0.5 \text{ with } p < 2 \textit{ atmospheres}$$

and $$x > 0.6 \quad „ \quad p \leqq 15 \quad „$$

As the steam acting in the cylinder of an engine is never so wet that $x = 0.6$, it follows that if the dryness-fraction is to remain constant, heat must be *imparted* to the steam during its expansion *behind* the piston, and *withdrawn* during the compression *before* the piston. If heat is imparted and withdrawn in this manner, the conditions described at the conclusion of 5) must arise.

14) *The internal work U* of a mixture of steam and water diminishes steadily during expansion with the dryness-fraction constant, for by Eq. 35 **Internal work U.**

$$A\,dU = dq + d(x\varrho)$$

and therefore here $A\,dU = dq + x\,d\varrho$

Just below Eq. 36[b] it was shewn that we may put

$$dq = Cdt.$$

by Eq. 27 $$\varrho = 575.4 - 0.791\,t,$$

so that $$A\,dU = Cdt - 0.791\,x\,dt$$

and $$A\frac{dU}{dt} = C - 0.791\,x.$$

As C and x can never exceed unity, the preceding value is always positive, wherefore, as dt and dp are both negative during expansion dU must also be negative, that is the internal work must decrease. During compression it increases correspondingly.

15) **V. Change of state with internal work constant.** If U is constant, we have, by Eq. 34 **Isodynamic curve for mixed steam.**

$$q + x\varrho = const. = q_1 + x_1\varrho_1$$

q, x, ϱ referring as before to the initial state and q_1, x_1, ϱ_1 to the terminal state. From this follows, for the equation of the isodynamic curve which corresponds to this particular change of state (see § 6, 27[b])

$$v_1 = x_1 u_1 + w$$

$$x_1 = \frac{q + x\varrho - q_1}{\varrho_1}$$

$$v_1 = w + \frac{q + x\varrho - q_1}{\varrho_1} u_1 \quad \ldots\ldots\ldots\ldots \quad (44)$$

because we can regard q_1, ϱ_1 and u_1 as functions of the terminal pressure p_1. As by the preceding

$$x_1 = \frac{q + x\varrho - q_1}{\varrho_1} = x + \frac{(q + x\varrho) - (q_1 + x\varrho_1)}{\varrho_1},$$

it follows that in the case of expansion

$x_1 > x$, because in expansion with a constant dryness-fraction (see 14) U must decrease and therefore we must have

$$q_1 + x\varrho_1 < q + x\varrho$$

Thus we see that isodynamic expansion (Plate 1, Fig. 4, Curve III) is accompanied by the evaporation of water. On account of this increase of x_1, u_1 increases less (for a given increase of v_1) and therefore p_1 decreases less, than would be the case if x_1 remained unchanged. This means that the isodynamic curve approaches the x axis, with increasing abscissæ v, less rapidly than the saturation curve does.

Heat-quantity dQ.

16) *The heat-quantity* dQ to be imparted to a mixture of steam and water for an infinitely small change of state with internal work constant, follows from Eq. 36

$$dQ = dq + d(\varrho x) + A p\, dv$$

as in this $dq + d(\varrho x) = A\, dU$ and in the present case $dU = 0$,

$$\therefore\; dQ = A p\, dv,$$

i. e. during expansion the whole heat supplied is converted into work and during compression the whole work performed is converted into heat.

Adiabatic curve for mixed steam.

17) **VI. Change of state without communication or withdrawal of heat.** If we put, by this hypothesis $Q = 0$ in Eq. 36

$$0 = dq + T\, d\left(\frac{x r}{T}\right),$$

and divide the equation by T, we have

$$\frac{dq}{T} + d\left(\frac{x r}{T}\right) = 0.$$

If we abbreviate thus,

$$\int_0^t \frac{dq}{T} = a \quad \text{and} \quad \frac{r}{T} = b,$$

the change of state in question will be characterized by the equation

$$a + b\, x = const. = a_1 + b_1\, x_1$$

in which a, b, x refer to the initial, and a_1, b_1, x_1 to the terminal states.

Eliminating x_1

$$x_1 = \frac{a + b\,x - a_1}{b_1}$$

and substituting this value in Eq 33, we have

$$v_1 = x_1\,u_1 + w = w + \frac{a + b\,x - a_1}{b_1}\,u_1 \quad . \; . \; . \; . \; . \; . \; . \quad (45)$$

as a_1, b_1 and u_1 are functions of p_1 Eq. 45 represents v_1, as a function of p_1. The curve which corresponds to such a change of state is called the *adiabatic curve* (Plate 1, Fig. 4, Curve V).

Table for constructing the adiabatic curve for mixed steam.

18) *The adiabatic curve of saturated steam* approaches the axis of x more rapidly than the saturation curve does. The adiabatic curve can be easily plotted by help of the table below. In it the values a and b up to 15 atmospheres were calculated by FLIEGNER*), while the values of u were recently determined afresh by PINZGER**) with $A = \frac{1}{424}$. The values from 16 to 30 atmospheres are due to ALTHANS***), A being taken as $= \frac{1}{436}$ (compare § 10, 11).

Table of the values of a, b, and u.

p in kg/□cm	a	b	u	p in kg/□cm	a	b	u	p in kg/□cm	a	b	u
0,10	0,1546	1,8041	14,8904	5,50	0,4528	1,1634	0,3423	12,25	0,5303	1,0268	0,1608
0,20	0,1984	1,6975	7,7354	5,75	0,4569	1,1559	0,3282	12,50	0,5323	1,0234	0,1578
0,30	0,2252	1,6344	5,2798	6,00	0,4609	1,1488	0,3154	12,75	0,5344	1,0199	0,1549
0,40	0,2448	1,5893	4,0279	6,25	0,4647	1,1419	0,3035	13,00	0,5364	1,0166	0,1521
0,50	0,2604	1,5541	3,2656	6,50	0,4683	1,1352	0,2924	13,25	0,5383	1,0133	0,1493
0,60	0,2734	1,5252	2,7510	6,75	0,4719	1,1288	0,2822	13,50	0,5403	1,0100	0,1467
0,70	0,2846	1,5007	2,3796	7,00	0,4753	1,1227	0,2727	13,75	0,5422	1,0068	0,1442
0,80	0,2944	1,4793	2,0984	7,25	0,4786	1,1167	0,2638	14,00	0,5440	1,0037	0,1418
0,90	0,3032	1,4605	1,8779	7,50	0,4819	1,1110	0,2555	14,25	0,5459	1,0006	0,1394
1,00	0,3111	1,4436	1,7002	7,75	0,4850	1,1054	0,2478	14,50	0,5477	0,9976	0,1371
1,25	0,3282	1,4076	1,3794	8,00	0,4881	1,1000	0,2405	14,75	0,5495	0,9946	0,1349
1,50	0,3424	1,3781	1,1621	8,25	0,4910	1,0947	0,2336	15,00	0,5512	0,9917	0,1328
1,75	0,3547	1,3530	1,0053	8,50	0,4939	1,0896	0,2271	16,00	0,5581	0,9804	0,1285
2,00	0,3655	1,3312	0,8867	8,75	0,4967	1,0847	0,2210	17,00	0,5645	0,9698	0,1213
2,25	0,3751	1,3119	0,7937	9,00	0,4995	1,0799	0,2152	18,00	0,5706	0,9598	0,1149
2,50	0,3839	1,2946	0,7188	9,25	0,5022	1,0752	0,2097	19,00	0,5765	0,9503	0,1092
2,75	0,3919	1,2789	0,6572	9,50	0,5048	1,0706	0,2045	20,00	0,5821	0,9413	0,1039
3,00	0,3993	1,2645	0,6056	9,75	0,5073	1,0662	0,1995	21,00	0,5875	0,9327	0,0993
3,25	0,4061	1,2513	0,5616	10,00	0,5099	1,0618	0,1948	22,00	0,5927	0,9244	0,0950
3,50	0,4125	1,2390	0,5238	10,25	0,5123	1,0576	0,1903	23,00	0,5977	0,9165	0,0911
3,75	0,4185	1,2275	0,4909	10,50	0,5147	1,0534	0,1860	24,00	0,6024	0,9090	0,0875
4,00	0,4242	1,2168	0,4620	10,75	0,5171	1,0494	0,1820	25,00	0,6071	0,9017	0,0844
4,25	0,4296	1,2067	0,4363	11,00	0,5194	1,0454	0,1781	26,00	0,6115	0,8948	0,0811
4,50	0,4347	1,1971	0,4135	11,25	0,5216	1,0415	0,1743	27,00	0,6159	0,8880	0,0782
4,75	0,4395	1,1880	0,3930	11,50	0,5239	1,0378	0,1707	28,00	0,6201	0,8815	0,0756
5,00	0,4442	1,1794	0,3744	11,75	0,5260	1,0340	0,1673	29,00	0,6242	0,8752	0,0731
5,25	0,4486	1,1712	0,3576	12,00	0,5282	1,0304	0,1640	30,00	0,6281	0,8691	0,0708

*) Civilingenieur 1874. P. 447 to 454.

**) MEHRTENS. Technische Mechanik. Berlin 1887. P. 833.

***) Zeitschrift für Berg-, Hütten- und Salinenwesen. 1875. P. 280.

Exact calculation of the external work L.

19) *The external work* L produced by 1 kilo of mixed steam in expanding from v to v_1 without receiving or giving up heat, while the dryness-fraction changes from x to x_1 and the pressure falls from p to p_1 is determined from Eq. 36

$$dQ = dq + d(x\varrho) + Ap\,dv = 0$$

$$Ap\,dv = -dq - d(x\varrho)\,,$$ this, when integrated gives

$$AL = q - q_1 + x\varrho - x_1\varrho_1$$

$$L = \frac{1}{A}(q - q_1 + x\varrho - x_1\varrho_1) \text{ kilogrammetres} \quad \ldots\ldots (46)$$

In this q and ϱ can be expressed in terms of p and q_1 ϱ_1 in terms of p_1, while the value of x_1 is given in terms of x by the equation worked out in 17)

$$x_1 = \frac{a + bx - a_1}{b_1}$$

Approximate Formula for the adiabatic curve of mixed steam.

20) From these formulæ we can compute the work produced by the expansion of saturated steam in nonconducting cylinders. But as a rule the initial state of the steam and the cut-off ε are the only data we have, so that it is desirable to be able to express the work of expansion as a direct function of the given quantities corresponding to the initial state of the steam, and of the cut-off ε, by means of an approximate formula. For this purpose RANKINE first introduced the equation $pv^n = p_1 v_1^n = constant$ for the adiabatic curve of mixed steam analogous to that for perfect gases $pv^\varkappa = p_1 v_1^\varkappa = const.$ In the equation for mixed steam however the index n is not, like $\varkappa$, a constant, but depends upon the cut-off ε, the initial pressure p, and the dryness-fraction x. It varies, nevertheless, so slightly with varying values of these quantities, that within certain limits, as the following numbers shew, it may be regarded for all practical purposes as a constant.

Index n.

21) *The value of the index* n is determined by GRASHOF*) from the formula

$$n = \alpha - \beta p + \gamma \log p + b \log \varepsilon$$

in which α, β, γ and b are coefficients depending upon x. As mean values GRASHOF gives, for

$x = 1$	$x = 0.9$	$x = 0.8$
$n = 1.1350$	1.1264	1.1158

*) F. GRASHOF. Theoretische Maschinenlehre. Leipzig 1875. P. 174.

ZEUNER*) says $n = 1.035 + 0.1\,x$
when x is between the limits 0.7 to 1.0.
From 1 to 12 atmospheres, RANKINE**) puts

$$n = 1.1111 = \frac{10}{9}$$

which corresponds with ZEUNER's formula when $x = 0.76$, i. e. when the steam contains about 25 % of moisture.

22) *The external work or work of expansion* L produced by the adiabatic expansion of mixed steam, as based on the preceding approximate formula, follows by Eq. 20 (P. 15)

Approximate calculation of the external work.

$$L = \frac{p\,v}{n-1}\left[1 - \left(\frac{v}{v_1}\right)^{n-1}\right] \text{ kilogrammetres;}$$

putting the cut-off $\frac{v}{v_1} = \varepsilon$

$$L = \frac{p\,v}{n-1}\left[1 - \varepsilon^{n-1}\right] \text{ kilogrammetres} \quad \dots\dots (46^a)$$

The terminal pressure p_1 follows from

$$p v^n = p_1 v_1^n$$

$$p_1 = p\left(\frac{v}{v_1}\right)^n = p\,\varepsilon^n \quad \dots\dots\dots\dots (47)$$

As u_1 is also determined by p_1 and can be taken from the Table on page 28 to 31, we have by Eq. 33

$$v_1 = \frac{v}{\varepsilon} = x_1 u_1 + w$$

and the terminal dryness-fraction

$$x_1 = \frac{\frac{v}{\varepsilon} - w}{u_1} \quad \dots\dots\dots\dots\dots (48)$$

The energy which must be exerted, *to compress* one kilo of the mixture is by Eq. 46ª

$$L = \frac{p\,v}{n-1}\left[\varepsilon^{n-1} - 1\right] \text{ kilogrammetres.} \quad \dots\dots (49)$$

23) In Fig. 4, Plate 1, the whole of the expansion curves are placed together for comparison. They are drawn on the

Comparison of the expansion curves of mixed steam.

*) G. ZEUNER. Mechanische Wärmetheorie. Leipzig 1877. S. 342.
**) J. W. M. RANKINE. The steam engine. London 1873. S. 392.

assumption that $x = 1$ i. e. that the steam is perfectly dry, and that just sufficient water is admitted to or withdrawn from the steam during its expansion according to the different curves, to keep it exactly at the saturation point. We see that, on the above assumptions, the saturation curve and the curve of constant internal work (isodynamic) nearly coincide. The adiabatic curve (drawn with ZEUNER's index $n = 1.135$) also lies very near them and approaches them the more closely the greater degree of moisture the steam possesses. RANKINE*), who usually represented the saturation curve for $x = 1$ by $p v^{\frac{17}{16}} = const.$ and the adiabatic curve of saturated steam by $p v^{\frac{10}{9}} = const.$ (according to ZEUNER's formula for steam of 25 % moisture), therefore recommends the use of the former curve for dry or nearly dry steam and the latter for moderately moist steam. But he says himself that the difference between the results of both curves is so small that in practice he advises the application of the former curve if we only have a table of squares at hand (on account of the 16th roots) and the latter if we only have a table of natural logarithms.

§ 12.

Determination of the degree of moisture of the steam.

Object.

1) The determination of the degree of moisture of the steam as it leaves the boiler is of great importance in fixing the capability of a boiler as well as in measuring the steam used in the engine. A whole series of the most diverse processes have been employed for this purpose, without, however, unexceptionably perfect scientific results being arrived at, although the numerical values obtained have in many cases sufficed for practical purposes and led to valuable inferences.

Division.

2) According to SEEMANN**), the different methods of investigation may be divided into

I. *Physical methods*

a) the calorimetrical process,

b) the weighing process,

c) the superheating process,

*) J. W. M. RANKINE. Useful rules and tables. London 1867. P. 290.

**) Zeitschrift deutscher Ingenieure 1885. P. 340.

II. *Chemical methods*
  d) the usual process
  e) ESCHER's process
  f) BRAUER's process, and
III. *Mechanical methods*, to which belongs
  g) MÖLLER's process.

3) **I. The physical methods** are all founded on the mechanical theory of heat. The oldest of them, the calorimetric method introduced by HIRN thirty years ago, is still the most used on account of the simplicity of its practical application. Physical Methods.

4) a. **The calorimetric method***) is based upon the view that the watery particles carried over by the steam permeate it throughout in the form of fine drops or bubbles, so that the steam and water together form a mixture of uniform composition. At a convenient place a branch, provided with a stop-cock, is fitted to the main steampipe and connected to a spiral terminating in a rose. The steam to be examined passes from the rose into a vessel partly filled with cold water and there condenses. The increase of weight and rise of temperature of the water in the vessel are measured and these determine the quantity of heat given off by a certain amount of the moist steam. From this quantity of heat the degree of moisture can then be calculated. The process.

5) To calculate the degree of moisture, let Calculation.

$M + m$ = the weight of the steam to be tested, consisting of M kilos of dry saturated steam and m kilos of water,

t = the temperature of the steam (centigrade),

N = the weight of water in the vessel before the experiment,

t_1 and t_2 = the temperatures of this water before and after the admission of the steam respectively,

λ, q and r = the corresponding heat quantities in § 10.

If the heat contained in the bodies thus mixed is to be the same before and after their mixture we must have

$$M\lambda + m\,q + N q_1 = (M + m + N)\, q_2$$

and referring to Eq. 31 and simplifying

$$m = \frac{(M + m)(\lambda - q_2) - N(q_2 - q_1)}{r}.$$

As only low temperatures are in question, HIRN puts $q_1 = t_1$ and $q_2 = t_2$ and writes

$$m = \frac{(M + m)(606.5 + 0.305\,t - t_2) - N(t_2 - t_1)}{606.5 + 0.305\,t - q} \quad \ldots . \quad (50)$$

*) Bulletin de la société industrielle de Mulhouse. 1869. P. 543.

From this we get the quantity of water contained in one kilo of the moist steam,

$$y = \frac{m}{M + m}.$$

Value of the calorimetric method.

6) *The sources of error of the calorimetric method* arise chiefly from the fact that Eq. 50 is only applicable up to the limiting value of $m = 0$, which corresponds to perfectly dry saturated steam. Therefore, in order to be certain that we have not to do with superheated steam, we must measure not only the temperature, but the pressure of the steam. Further — the quantity of water in the vessel must not be too small; HIRN and HALLAUER take 30 to 50 litres. The most scrupulous exactness must be observed in taking the weights and the temperatures, the latter must be observed to a tenth of a degree at least. There are also losses of heat by convection and radiation, and by the pipes and vessels taking up heat, all of which must be corrected for. Lastly this method, like all those which depend upon what may be called "sampling", lies under the objection that only a small quantity of steam can be tested, of which we are not certain that its composition really corresponds with the mixture of the main boiler steam. HIRN and HALLAUER found in their experiments that the degree of moisture oscillated between 2 % and 5 % and varied considerably in every case. STAHLSCHMIDT*) calculated degrees of moisture from 0 to 3 % in his experiments under the same working conditions as above, so that he did not apply this method when testing the boilers as the Düsseldorf Exhibition in 1880 as he had intended. LORING and EMERY**) in their experiments on the steamer "Gallatin", to be further described in the sequel, got 0.05 to 4.8 % of moisture with steam at 4.22 to 4.93 kilos per □cm pressure. LINDE***), by *continually* condensing his test steam, by means of a small surface condenser, obtained nearly dry steam, while the condensation process of the engine shewed 7 to 8 % of moisture. So that steam may be in fact more moist than it appears to be by the calorimetric test.

Process.

7) b. **The weighing method** is of a strictly physical character and was first employed by GUZZI****) and KNIGHT†). Both make use of a

*) Untersuchungen von Dampfmaschinen etc. der Gewerbe-Ausstellung in Düsseldorf 1880. Aachen 1881. P. 13.

**) Engineering. 1876. I. P. 125.

***) Bericht über die 5. Versammlung des Verbandes der Dampfkessel-Ueberwachungs-Vereine. München 1877. P. 43.

****) Revue industrielle 1878. P. 102.

†) Journal of the Franklin Institute 1877. P. 358.

copper globe as a measuring vessel which is placed in a receiver connected to the main steam pipe, and after being filled is taken out and weighed. GUZZI connects this receiver to a branch off the main steam pipe and blows it through before the experiment. KNIGHT inserts his apparatus bodily into the main steam pipe and arranges the globe so that it can be opened and filled, or the water blown out of it while it remains in the steam pipe. Another new process of this sort was introduced by CARIO, but is less to be recommended than the older ones of GUZZI and KNIGHT.

8) To calculate the moisture, let **Calculation.**

V = the volume in cubic metres of the mixture of $M + m$ kilos,

γ = the density of the dry saturated steam,

γ_w = " " " " water,

the other designations remaining as in 5). We then get

$$\frac{M}{\gamma} + \frac{m}{\gamma_w} = V$$

$$m = \frac{(M+m) - V\gamma}{1 - \frac{\gamma}{\gamma_w}} \quad \ldots\ldots\ldots\ldots \quad (51)$$

The volume of the water in the mixture is usually neglected, so that

$$M = V\gamma$$

and

$$m = (M+m) - V\gamma \quad \ldots\ldots\ldots\ldots \quad (51^a)$$

The weight of water m thus comes out as the difference between the weight of $(M+m)$ as obtained by experiment and the weight of an equal volume of dry steam. Therefore we must measure the weight and volume of the mixture to be tested as well as its pressure or temperature. The quantity of water in one kilo of the moist steam is as before

$$y = \frac{m}{M+m} = 1 - \frac{V\gamma}{M+m}$$

For want of sufficient experiments it cannot yet be said how far the weighing method may be regarded as of practical value.

9) c. **The superheating method** is based upon the following idea. A certain weight of moist steam is enclosed in a cylinder, in which a steam-tight piston can be moved at will. Heat can be communicated to the apparatus in such a manner that while the piston is moved forwards (or outwards) the water in the moist steam evaporates at a constant temperature. So long as any water is contained in the mixture, the steam remains saturated and the pressure constant, but if we continue the expansion and the communication of heat far enough we shall arrive at **Process.**

a point where the dry steam becomes *superheated*, the sign of which is that the pressure begins to fall. BROCQ's*) apparatus is constructed on this basis. It consists of a brass cylinder surrounded by a jacket which serves to heat the cylinder and is connected so that the main steam passes through it. The cylinder is provided with two slide valves which can be simultaneously opened or shut from outside. By means of the two slide valves the steam is allowed to pass through the cylinder until the latter is properly heated up, or in other words until it has the same temperature as the steam. The slide-valves are then closed, and the piston is moved slowly outwards until an extremely sensitive steamgauge fitted to the cylinder and connected with an electric bell announces a fall in the pressure.

Calculation. 10) *The calculation of the degree of moisture*, after the volume swept by the piston has been observed, the designations in 5 and 8 and the values of x, w and u in § 11, 2 being borne in mind, now takes the following form.

If $(M+m)$ kilos of moist steam of a dryness fraction x are admitted to the cylinder, we have the volume V_1 of the mixture at the moment when the superheating begins

$$V_1 = (M+m)(w+u),$$

because the specific volume of the steam now become dry and just saturated is $w+u$; whereas the initial volume was by Eq. 33

$$V = (M+m)(w+xu)$$

Subtracting the latter equation from the former, we get y the "specific quantity of water" or weight of water in 1 kilo of the moist steam,

$$y = 1 - x = \frac{V_1 - V}{(M+m)u} = \frac{V_1}{(M+m)u}\left(1 - \frac{V}{V_1}\right) = \frac{1-\varepsilon}{\gamma u} \quad . \quad (52)$$

in which $\varepsilon = \frac{V}{V_1}$ is the cut-off, and $\gamma = \frac{M+m}{V_1}$ the density, or weight in kilos of a cubic metre of saturated steam of the observed initial pressure, the corresponding value of u for which is given in the steam table on pages 28 to 31. As $\frac{1}{\gamma} = w + u$ and, neglecting $w = 0.001$, we may put $\frac{1}{\gamma} = u$, we get approximately

$$y = 1 - \varepsilon = \frac{V_1 - V}{V_1} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (52^a)$$

i. e. the initial degree of moisture is directly proportional to the increase of volume up to the point of saturation, in other words to the point where superheating begins.

*) Revue industrielle. 1881. P. 334.

11) *The sources of error of this method* arise from the difficulty of getting steam of the same moisture into the little cylinder as in the main steam-pipe and of extending the experiment over a sufficiently long time. The piston must also be moved very slowly because steam is a bad heat-conductor and the temperature in the cylinder and jacket may vary, thus rendering the change of state no longer isothermal. Value of the superheating-method.

12) **II. The Chemical Methods** have the advantage over the physical ones that they can be applied with very simple apparatus. They are founded on the assumption that pure steam is free from any admixture present in the boiler water and that, on the other hand, the water carried over with the steam contains exactly the same ingredients as the highest part of the water in the boiler. Chemical Methods.

13) d. **The usual chemical method** as employed at the Düsseldorf Exhibition in 1880*) consists *imprimis* in dissolving 20 to 25 kilos of sulphate of soda (according to the size of the boiler) in the boiler water. From time to time during the working a certain quantity of water (at Düsseldorf $^1/_2$ kilo) is drawn from the boiler and simultaneously a certain quantity of steam from the main steam-pipe. The steam can be best taken off by means of a small pipe inserted into and across the main steam-pipe and provided throughout its length with a number of small holes whose direction is opposite to the current of steam. The steam on passing from this small pipe is condensed in a copper worm, and equal quantities of the resulting condensed water and of the boiler water are reserved for examination. At Düsseldorf this process was repeated ten times, after which the ten samples of boiler water were mixed together in one vessel and the ten samples of the water condensed from the steam in another. The percentage of sulphuric acid (derived of course from the sulphate of soda put into the boiler) in the two kinds of water then gave a means of determining the degree of moisture of the steam. For this purpose equal quantities of chloride of barium were added to the two waters and the resulting precipitate of sulphate of baryta weighed. Process.

14) *The calculation of the degree of moisture* is very simple. If the analysis shews k kilos of dissolved salts (or as above, sulphate of baryta) per kilo of the boiler water, we have, for the saline contents d of the condensed water Calculation.

$$d = y\,k$$

*) Die Untersuchungen von Dampfmaschinen etc. der Gewerbe-Ausstellung in Düsseldorf 1880. Aachen 1881. P. 13.

if y represents the degree of moisture of the steam (i. e. the weight of water in a kilo of steam).

$$\therefore y = \frac{d}{k} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (53)$$

We may remark that in the Düsseldorf experiments*) the moisture in steam of 5 atmospheres (75 ℔s) working pressure, which on leaving the boiler escaped into the air, varied between 0.21 and 9 % but in steam on its way to the engine, between 0.56 and 1.1 %. Mr. EICKENRODT of the Imperial German Navy in some most unexceptionable experiments on board the cruiser "Albatross" found it to be 1.65 % in steam of 2 atmospheres or 30 ℔s working pressure.

Value of the usual chemical method.

15) *The sources of error of this method* arise principally from the difficulty of getting a sample of steam of the same composition as that of the steam in the main steam pipe, a fault common to all methods in which only a *small* quantity of steam is taken for examination (see 6). And in order to get water of the same strength (of solution) as the water in the boiler and not a mixture of steam and water, we must draw it off into a steam-tight vessel screwed on to a gauge cock or a separate cock for the purpose. This cock must be closed after the vessel is full and the latter not disconnected from the boiler until it has cooled down. A still better arrangement is to connect the vessel in such a manner with the boiler that the boiler water can circulate through it for some time before it is disconnected. Sometimes the water gauge glass itself may be used for the purpose.

Process.

16) e. **ESCHER's Method****) starts from the assumption that when a boiler is first set to work, both the water in the boiler and the water with which it is to be fed contain the same percentage of matter in solution (whatever it may be) so that from time to time as the boiler continues to be fed with this water and to give off pure steam, the degree of concentration of the solution in the boiler rises. The law of this increase of concentration, being observed, gives a measure of the quantity of water carried over by the steam. This process has the advantage of dispensing with the repeated withdrawal of small samples of steam.

Calculation.

17) *The calculation of the degree of moisture* is as follows. If s is the weight of dissolved salts per kilo of the feedwater and k the corresponding weight for the boiler water, we can regard

*) Die Untersuchungen von Dampfmaschinen etc. der Gewerbe-Ausstellung in Düsseldorf 1880. Aachen 1881. Pp. 20, 25 and 26.

**) Civilingenieur 1879. P. 51.

s as constant, whereas k varies with the time. Let k then represent the weight of salt per kilo in the boiler water at the end of the time t reckoned from when the boiler was newly filled and set to work, D the weight of feedwater per hour, and K the total weight of water in the boiler. The boiler is to be assumed to be in a "permanent state" of working, i. e. the weight of feed supplied and steam delivered in unit time are to be considered as equal, or in practical language the water is to be just steady in the glass and neither to rise nor fall. Accordingly we have, as the quantity of salt put into the boiler with the feed water in the time dt

$$m_1 = D\,dt\,s$$

In the same time the steam, the weight of water carried over per kilo of which is y, takes out of the boiler the quantity of salt

$$m_2 = D\,y\,dt\,k\,,$$

so that the increase of saltness in the boiler water during the time dt is

$$m_1 - m_2 = D\,(s\,dt - y\,k\,dt) = K\,d\,k\,,$$

$d\,k$ representing the increase of weight of salt per kilo of water in the boiler.

If we put the ratio $$\frac{K}{D} = a$$

we get the differential equation

$$d\,k + \frac{y}{a}\,k\,dt - \frac{s}{a}\,dt = 0$$

the general integral of which is

$$k = \frac{s}{y} + \frac{C}{e^{\frac{y}{a}t}}\,.$$

To determine the constant C, we see that for

$$t = 0\ ,\ k = s = \frac{s}{y} + C\,,$$

so that $$C = -\frac{s}{y}\,(1 - y)$$

and the complete integral

$$k = \frac{s}{y}\left(1 - \frac{1-y}{e^{\frac{y}{a}t}}\right) \qquad \ldots\ldots\ldots\ldots \quad (54)$$

This equation shews that the coefficient of concentration k approaches assymptotically the limit

$$k_{max} = \frac{s}{y}\,,$$ which it reaches

when $$t = \infty\,.$$

The magnitude of y can only be determined from the equation for k by gradual approximation. The duration of the period of working, at the end of which the value of k (weight of salt per kilo) of the boiler water has been observed is t, and s is known beforehand. As a first approximate value we have

$$y = \frac{s}{k}$$

Substituting this in the above expression, we get a second approximate value, and so on.

Value of Escher's Method.

18) *The sources of error of this method* lie in the practical impossibility of keeping the feed and production of steam exactly balanced, as is assumed to be the case. This method further disregards the deposition of crystals of salt which are certainly to be expected as the solution becomes more and more concentrated.

Process.

19) f. **In BRAUER's Method***), as in the usual method, any salt, say common salt (chloride of sodium), is dissolved in the boiler water. In the case of marine boilers, it is sufficient to fill them with sea-water. If the boiler is now fed with pure (distilled) water, the saline contents of the boiler water must diminish if the steam carries over any water (i. e. brine) with it. This gradual reduction of the strength of the brine (the rate of steam generation being known) bears a certain relation to the degree of moisture of the steam. By testing the water with a salinometer at the beginning of the process, repeating the salinometer tests from time to time and accurately measuring the feed water supplied during the experiment, the average degree of moisture of the steam generated between any two of the salinometer tests may be calculated, care being taken that there is exactly the same height of water in the gauge-glass at every test.

Calculation.

20) *To calculate the degree of moisture*, let

s = the variable weight of salt contained in 1 kilo of the brine,

s_1 and s_2, the values of s at the beginning and end of the experiment respectively,

G the total weight of brine in the boiler which, with a weak solution and constant volume, may be regarded as constant.

Then $S = Gs$, the total weight of salt contained in the brine. Consequently $dS = G\,ds$.

If we call y the weight of water in 1 kilo of the mixture of steam and water, there corresponds to this y a weight of salt

*) Wochenschrift des Vereines deutscher Ingenieure 1883. P. 158.

of sy kilos. Further, if W is the weight of water evaporated since the last salinometer test and therefore dW the weight evaporated in an infinitely short period, the latter carries over with it as moisture $y\,dW$ kilos of brine and consequently $sy\,dW$ kilos of salt. This quantity of salt evidently represents the decrement of the total weight of salt in the boiler corresponding to the evaporation of dW, that is dS.

So that $dS = sy\,dW$. From this and the Equation above, it follows that

$$G\,ds = sy\,dW$$

$$\frac{ds}{s} = \frac{y}{G}\,dW$$

$$\int_{s_2}^{s_1} \frac{ds}{s} = \frac{y}{G} \int dW$$

$$ln\frac{s_1}{s_2} = \frac{y}{G}W$$

$$v = \frac{G}{W}\,ln\frac{s_1}{s_2} \qquad \qquad (55)$$

21) *The sources of error of Brauer's Method* are in general the same as those of ESCHER's, but BRAUER's has the advantage of weakening the brine, thus reducing the tendency to form scale. Its chief merit however, is that we always have it in our power to get a thoroughly uniform solution in the boiler by using considerable quantities of an easily soluble salt. But of course there must be nothing in the feedwater which would precipitate the salt we put into the boiler, for instance if common salt is used as above described, the feedwater must not contain chloride of barium. **Value of Brauer's Method.**

22) *The advantage of the two latter methods* over all others is that the observations required are all confined to the state of affairs in the *boiler*, the steam in the pipes not being considered at all. So that it is quite indifferent (for the purposes of the experiment), what relation may there exist between the steam and any water it contains, either condensed or carried over. This is the more important as the molecular character of the steam is withdrawn from direct observation and as, on the other hand, it is still an open question whether the water is carried over steadily or by gulps. BUNTE's*) first application of BRAUER's method on the occasion of the portable-engine competition at Berlin in 1883, when 1 % of common salt was mixed with the boiler water, gave no positive results, as only infini- **Advantage of the two latter methods.**

*) Civilingenieur 1884. P. 207.

tesimal diminutions of the concentration could be shewn, pointing to nearly dry steam, which is certainly not peculiar to boilers of the portable type. BUNTE however has in later experiments found this process useful, as the inventor writes me, for boilers having *one* water space, in which it may be assumed that the concentration of the solution is uniform throughout.

Mechanical Methods.

23) **III. The mechanical methods** aim at clearing the steam from the water it carries over by the introduction of separators. It appears from the most recent experience that this method is the likeliest one to lead to an exact determination of the moisture of steam, assuming that the apparatus is good.

Moeller's Steam-filter.

24) g. **MOELLER's process***) is an attempt to separate the watery particles as well any solid bodies from the steam by means of a steam-filter arranged for the purpose. The steam-filter consists of a vertical cylindrical shell, cleaded with nonconducting material, into which the steam enters by a horizontal branch at its lower end. A number of tubes, closed at the bottom and open at the top are suspended in the shell from a tube-plate. These tubes are composed of a peculiarly arranged basket-like web made of brass or iron and are covered on their outer surfaces with some very porous filtering substance. The mixture of steam and water passes from the outside to the inside of the tubes and the filtered steam escapes at their open upper ends to the chamber above the tube-plate. The water adheres to the filtering substance on the outside of the tubes, drips off, and collects at the bottom of the shell, whence it can be blown out through a cock. The dry steam passes off at the upper end of the shell and through the main steam pipe to the engine.

Value of Moeller's Steam-filter.

25) The usefulness of MOELLER's steam-filter was clearly shewn in some experiments carried out in London by CLARK**) in order to determine the loss of heat from steampipes, both bare, and cleaded with different nonconducting materials. In this case the steam was dried in one of these filters before entering the pipes to be experimented on. The filter was also used by FFRD. FISCHER***) in some evaporating experiments at Hainholz near Hanover, in which however the steam carried over no water. MOELLER's filter has also been employed at the Imperial Dockyard at Wilhelmshaven in some experiments to determine the evaporative value of different sorts of coal, and worked very well, even during intentionally-caused priming.

*) Zeitschrift des Vereines deutscher Ingenieure 1888. P. 398.

**) The Engineer. 1884. P. 66.

***) Zeitschrift des Vereines deutscher Ingenieure 1886. P. 46.

Unfortunately the inventor declares the fitting of his apparatus to marine engines to be impracticable because the filter would assume too large dimensions and a considerable weight, in order to suit the rapid production of steam of marine boilers. It has not shewn itself useful as a grease-separator for marine engines, for although it separated 30 kilos of greasy water out of 450 kilos of steam, it is not much use for this purpose, as it was found extremely difficult to separate the grease from the water. Consequently the whole of the greasy water had to be blown out, thus causing a sacrifice (as the above figures shew) of about 7 % of the condensed feed-water.

26) According to all the preceding, BRAUER's process would appear to recommend itself the most for a practical investigation of the moisture of steam, with the high working pressures of modern marine engines and extending over a lengthened run. It would be necessary for this purpose to run all the boilers up with sea-water at flood tide before the experiment, so as to get the same saltness in all, and then to feed exclusively with fresh water, that is, to use condensed or distilled supplementary feed. The arithmetical mean of the saltnesses of all the boilers at each test would be taken for determining the value of s_2 (Eq. 55). This is of course easily done with the help of the ordinary salinometer. **Best method for marine boilers.**

27) *Moist steam is dried*, as we may here shortly mention, either mechanically by means of separators, to be described further on with superheaters, or by throttling the steam in the steam-pipe, which process will be more closely gone into under the heading of water tube boilers; or lastly, by taking the steam through superheaters on its way from the boiler to the engine. **Drying moist steam.**

§ 13.

Superheated Steam.

1) **I. Equations of state.** *The equation of state of superheated steam* (see § 6, 19) i. e. the relation between its absolute pressure p in kilos per □cm., its specific volume v in cbm., and its absolute temperature T, has, according to ZEUNER,*) the following form: **Equation of state.**

*) Zeitschrift des Vereines deutscher Ingenieure 1866. P. 1.

$$pv + Cp^{\frac{\varkappa-1}{\varkappa}} = RT \text{ mk.}, \quad \ldots\ldots\ldots\ldots \quad (56)$$

in which

$$R = 0.0050933$$
$$C = 0.1925$$
$$\varkappa = 1.333 = \frac{4}{3}$$
$$\frac{\varkappa - 1}{\varkappa} = \frac{1}{4}$$

Eq. 56 may also be written thus

$$pv = RT - Cp^{\frac{\varkappa-1}{\varkappa}}$$

$$pv = R\left(T - \frac{C}{R}p^{\frac{\varkappa-1}{\varkappa}}\right) = R\left(T - \frac{C}{R}p^{\frac{1}{4}}\right)$$

and after substituting the above constants

$$pv = 0.0050933\left(T - 37.79475\sqrt[4]{p}\right) \text{ mk.} \quad \ldots\ldots \quad (56^a)$$

Heat of the steam I.

2) *The heat of the steam I* of 1 kilo of superheated steam is calculated from the formula

$$I = I_0 + \frac{A}{\varkappa - 1}pv \text{ thermal units} \quad \ldots\ldots \quad (57)$$

in which the constant I_0 has the following values for

$$p \geqq 0.5 \text{ kg}/\square\text{cm}, \; t > 80^0, \; I_0 = 476$$
$$p < 0.5 \;\;\text{,,} \;\;\;\; \text{,,} \;\;, \; t \leqq 80^0, \; I_0 = 478$$
$$p < 0.1 \;\;\text{,,} \;\;\;\; \text{,,} \;\;, \; t \leqq 45^0, \; I_0 = 480$$

Total heat λ.

3) *The total heat λ* required for the generation of 1 kilo of superheated steam of the pressure p kilos per $\square$cm. and the temperature T from water at 0^0 is found thus

$$\lambda = I_0 + c_p\left(T - \frac{C}{R}p^{\frac{1}{4}}\right) \quad \ldots\ldots\ldots\ldots \quad (58)$$

In this equation, besides the above constants, we have c_p the specific heat of superheated steam at constant pressure = 0.4805

$$\lambda = I_0 + 0.4805\left(T - 37.79475\sqrt[4]{p}\right) \text{ thermal units} \quad (58^a)$$

Adiabatic Curve for Superheated Steam.

4) **II. Changes of state.** *Change of state without communication or withdrawal of heat. (Adiabatic Curve.)* The external work which 1 kilo of superheated steam performs in passing from an initial state p and v into a terminal state p_1 and v_1, heat being neither imparted nor withdrawn, follows from Eq. 20 P. 15 as

$$L = \frac{pv}{\varkappa - 1}\left[1 - \left(\frac{v}{v_1}\right)^{\varkappa - 1}\right] \text{ mk.} \quad \ldots\ldots\ldots \quad (59)$$

or, because for superheated steam $\varkappa = 1.333$,

$$L = \frac{pv}{0.333}\left[1 - \left(\frac{v}{v_1}\right)^{0,333}\right] \text{ mk.}$$

If we call the ratio of cut off $\frac{v}{v_1}$, ε, it follows

$$L = 3pv\left[1 - \sqrt[3]{\varepsilon}\right] \text{ mk.} \quad \ldots\ldots\ldots\ldots \quad (59^a)$$

5) The foregoing equation is only correct so long as the superheated steam remains superheated during the expansion. At a high ratio of expansion the superheated steam passes over into the saturated state and then expands no longer according to the formula

Expansion of steam in the superheated and saturated conditions.

$$pv^{1,333} = p_1 v_1^{1,333},$$

but from the moment in which the steam enters the saturated condition and as it contains no water (i. e. $x = 1$) the expansion continues according to the formula

$$pv^{1,065} = 1.76133,$$

if p is taken in kilos per □cm (see Eq. 42ª, P. 38).

We may remark that it follows from the exponent $\varkappa = 1.333$ of superheated steam being higher than the exponent $n = 1.135$ of saturated steam, that the adiabatic curve of superheated steam approaches the axis of abscissæ more rapidly than that of saturated steam and is therefore more like the adiabatic curve of the perfect gases drawn on Plate 1, Fig. 4.

6) *The ratio of cut-off* ε, at which superheated steam goes over into a state of saturation at the end of the expansion, can be easily calculated from the foregoing. At the limiting state p_0 and v_0, we must have

Transition of expanding steam from the superheated to the saturated state.

$$p\,v^{1,333} = p_0\,v_0^{1,333}$$

and

$$p_0\,v_0^{1,065} = 1.76133$$

whence follows, by combination

$$\varepsilon = \frac{v}{v_0} = \left(\frac{1.76133}{p\,v^{1,065}}\right)^{\frac{1}{1,333 - 1,065}}$$

$$\varepsilon = \frac{v}{v_0} = \left(\frac{1.76133}{p\,v^{1,065}}\right)^{3,7313} \quad \ldots\ldots\ldots\ldots \quad (60)$$

If the cut-off is later than the limiting value found by Eq. 60, the expansion-work of the superheated steam is calculated by Eq. 59[a], but if it is earlier than this then the expansion-work up to the limiting volume v_0 must be taken from Eq. 59[a], and from thence to the end of the expansion from Eq. 46[a], P. 43.

Increase of temperature required to keep the steam superheated during expansion.

7) *The increase of temperature* which saturated steam of the pressure p in "old Atmospheres" must experience in order to be sufficiently superheated that it only returns to the saturated state at the *end* of an adiabatic expansion whose ratio of cut-off is ε, is calculated, according to GRASHOF*) from the formula

$$t = 335.24\, p^{0,0607} \left[\left(\frac{1}{\varepsilon}\right)^{0,2524} - 1\right] \quad \ldots\ldots\ldots (61)$$

on which the following table is based.

Table of the degrees of rise *(t)* of temperature, required to keep the steam superheated.

ratio of cut-off ε	Absolute pressure in "old Atmospheres" $p = 1$ t	$p = 2$ t	$p = 3$ t	$p = 4$ t	$p = 6$ t	$p = 8$ t	$p = 10$ t
1	2	3	4	5	6	7	8
0,10	264,2	275,6	282,4	287,4	294,6	299,7	303,8
0,15	205,9	214,7	220,1	224,0	229,6	233,6	236,8
0,20	168,0	175,2	179,6	182,7	187,3	190,6	193,2
0,25	140,4	146,5	150,1	152,8	156,6	159,3	161,5
0,30	119,0	124,2	127,3	129,5	132,7	135,1	136,9
0,40	87,2	91,0	93,2	94,9	97,2	99,0	100,3
0,50	64,1	66,9	68,5	69,7	71,5	72,7	73,7
0,60	46,1	48,1	49,3	50,2	51,4	52,3	53,0
0,80	19,4	20,2	20,7	21,1	21,6	22,0	22,3

Isodynamic Curve for superheated steam.

8) *Change of state with internal work constant. (Isodynamic curve.)* If the internal work or its heatvalue (i. e. for steam, the "heat of the steam I") is to remain constant, it follows by Eq. 57 that for 1 kilo of superheated steam passing from the initial state p, v to the terminal state p_1, v_1

$$I = I_0 + \frac{A}{\varkappa - 1} pv = I_0 + \frac{A}{\varkappa - 1} p_1 v_1$$

$$\therefore\ p_1 v_1 = p v \quad \text{mk.} \quad \ldots\ldots\ldots (62)$$

So that the isodynamic curve for superheated steam is, like that for the perfect gases, on equilateral hyperbola.

*) F. GRASHOF. Theoretische Maschinenlehre, Leipzig 1875. Vol. I. P. 212.

9) *Change of state with constant temperature. (Isothermal curve.)* With $p\,v$ referring to the initial and $p_1\,v_1$ to the terminal state, and as T is constant, we get by Eq. 56 **Isothermal curve for superheated steam.**

$$pv = R\,T - C p^{\frac{\varkappa-1}{\varkappa}}$$

$$p_1\,v_1 = R\,T - C p_1^{\frac{\varkappa-1}{\varkappa}}$$

and hence by subtraction

$$p_1\,v_1 = pv + C\left(p^{\frac{\varkappa-1}{\varkappa}} - p_1^{\frac{\varkappa-1}{\varkappa}}\right) \text{ mk. } \ldots\ldots (63)$$

On comparing this equation with that of the isodynamic curve, Eq. 62, we see that from any particular point the isodynamic curve for superheated steam approaches the axis of abscissæ more rapidly than the isothermal curve.

10) *Change of state with constant pressure.* The total heat necessary for the generation of M kilos of superheated steam of the temperature T from water of 0^0 is by Eq. 58 **Change of state with constant pressure.**

$$Q = M\lambda = M\left[I_0 + c_p\left(T - \frac{C}{R}p^{\frac{\varkappa-1}{\varkappa}}\right)\right] \text{ thermal units } (64)$$

and the work produced is, by Eq. 56:

$$L = Mpv = M\left[R\,T - C p^{\frac{\varkappa-1}{\varkappa}}\right] \text{ mk.}$$

If, at the same pressure p, the temperature of the steam is a different one T_1, and if the energy produced (and therefore the total volume of the steam generated) is to be the same as before, we have for M_1, the new weight of steam, the following relation

$$M\left(T - \frac{C}{R}p^{\frac{\varkappa-1}{\varkappa}}\right) = M_1\left(T_1 - \frac{C}{R}p^{\frac{\varkappa-1}{\varkappa}}\right) \text{ kilos } \ldots (65)$$

So that if we wish to substitute for steam of pressure p and temperature T other steam of equal pressure but of the temperature T_1, the weight M_1 of the latter is calculated by Eq. 65 and the heat quantity Q_1 by Eq. 64.

11) *Example.* M kilos of saturated steam of 5 "old Atmospheres" pressure, for which $t = 152.22^0$, are given and an equal volume of superheated steam at the temperature $t_1 = 200^0$ is to be produced. The weight of water required is given by Eq. 65 as **Example.**

$$M_1 = 0.8852\,M \text{ kilos.}$$

The total heat of the saturated steam is taken from the steam table on P. 29,

$$Q = 653.05\,M \quad \text{thermal units}$$

and that for the superheated steam from Eq. 64 viz.

$$Q_1 = 676.00\,M_1 \quad \text{thermal units;}$$

whence follows by the known relation of M to M_1

$$Q_1 = 0.9163\,Q \quad \text{thermal units.}$$

Thus it is evident that, other things being equal, *less* heat is required to generate a certain volume of superheated than of saturated steam.

Change of state with constant volume.

12) *Change of state with constant volume.* If heat is communicated to 1 kilo of steam which is kept at constant volume, only the "heat of the steam" is altered, as no external work is either expended or produced. So that for the initial state pv and the terminal state $p_1 v_1$, we have

$$Q = I_1 - I,$$

or referring to Eq. 57,

$$Q = \frac{A}{\varkappa - 1}(p_1 - p)\,v \quad \text{thermal units} \quad \ldots\ldots\ (66)$$

which shews the relation between the terminal pressure p_1 and the imparted heat-quantity Q.

Advantages of superheated steam.

13) **III. Advantages of superheated steam.** It follows from the foregoing and also from what is stated in § 8 and § 11 that superheated steam possesses the following advantages over steam which is saturated and permeated with watery particles.

No reduction of pressure with fall of temperature.

14) a. Superheated steam can experience a fall of temperature without a loss of pressure, so that it enters the cylinder of the engine at the same pressure at which it leaves the boiler, whereas saturated steam in marine engines loses as a rule about 0.5 kilos per □cm (say about 7 ℔s per □″) on its way from the boiler to the cylinder.

Saving of fuel.

15) b. With the same cut-off, the weight of superheated steam used is, in consequence of its lower density, less than that of saturated steam. Or, what is the same thing, the generation of a certain volume of superheated steam requires less feed-water and by 11) less fuel than an equal volume of saturated steam. For two steam-engines of the same size working at the same revolutions and *carrying the steam the whole of the stroke*, the one however using superheated steam and the other saturated steam as specified in 11) we see at once from the proportion of the heat quantities there calculated, that there is an advantage of 9.13 % of the fuel in favour of the engine

using superheated steam. But if the engines work *expansively*, this advantage of the saturated steam is diminished, because its adiabatic curve falls more rapidly than that of the saturated steam (see 5) and therefore its work of expansion is less.

16) In jacketed cylinders, superheated steam effects an increase of the mean temperature of the cylinder wall, by which the condensation during admission is diminished and re-evaporation accelerated, so that this is generally completed during the expansion. By using superheated steam in the jacket as well as in the cylinder, the mean temperature of the cylinder wall can be kept above the saturation temperature of the steam. In the latter case, the superheated steam behaves like a perfect gas and the losses of heat or steam are almost zero. Superheated steam however only develops this quality when its temperature is sufficiently high. For instance, to get this result with steam of 1 to 2 atmos. pressure, and consequently 120° to 133° C. temperature of saturation, we must heat it to about 180° or 200° C. But even thus highly superheated steam of 2 atmos. pressure when cut off before half stroke returns to its saturated condition during expansion, and if it is only heated to 160° it will do this with any earlier cut off than 70 %. Steam of 3 atmos. pressure superheated to 170° becomes saturated with less than 80 % cut off; and at four atmos., superheated to 180°, with less than 70 %. We must remark here that it does not appear to be advantageous to use superheated steam in unjacketed cylinders, for according to calculations of GRASHOF's*) from some experiments made by HIRN in 1875 the steam gave up 14 thermal units to the cylinder wall per stroke because the temperature of the latter was lower than the terminal temperature of the steam. The engine used in the experiments was an unjacketed "common-jet" one, using steam of 2.3 kilos per □cm superheated to 223° and cutting off at 45 %.

Reduction in the losses of steam.

17) **IV. Practical limit to superheating.** From the foregoing we see that the superheating must be driven as high as possible in order to obtain economical working. But the experience of years has shewn that it is not advisable to heat steam of 1 to 2 atmos. pressure to more than 160° i. e., to superheat it more than 40° and 30° respectively. When superheated above this limit the steam burns the packing, volatilises the lubricants used for cylinders and slide faces, and causes their hard residue to destroy the smoothness of the working surfaces,

Destruction of the smooth surfaces of the cylinders.

*) Zeitschrift des Vereines deutscher Ingenieure 1883, P. 176.

thus producing a considerable increase of friction. For this reason the steam in trunk-engines is never superheated more than 10° on account of their large rubbing surface.

Coppering of the cylinder surfaces.

18) In several cases portions of the slidefaces, sliderods, &c. have been attacked by low pressure steam superheated to 200° in such a manner that they seemed to have been burnt in a fire. It has even occurred that the backs of the slides, the slide-faces, and parts of the inside of the cylinders have appeared to be electroplated with copper. A feasible explanation of this phenomenon is that the saline water carried over from the boiler into the superheater is there decomposed by the high temperature and hydrochloric acid liberated. This, in the first place, attacks the copper steam pipe, forming chloride of copper which is again decomposed on coming into contact with iron and covers it with a thin film of copper.

Too high superheating to be avoided.

19) In order to avoid too high temperatures of superheating it is usual to carry only a portion of the steam through the super-heater and afterwards to mix this portion with moist steam, thus forming a mixture as described in § 8, 9; more rarely a small stream of water is run from a lubricator on to the slide face in order to cool the superheated steam. In many cases the heating surfaces of superheaters have been originally designed so small that a high superheating temperature could not possibly be reached, and generally speaking the arrangements called "superheaters" have not deserved the name at all, because they only acted as steam driers.

Results of Experiments.

20) Under these circumstances the anticipated saving of fuel remained in practice, as was natural, far behind the results of some experiments, shewn in the following table.

Table shewing the saving of fuel obtained in some experiments with superheated steam.

Experimenter	Date	Place	Subject	Pressure in kilos per cm.	Saturation temperature	Degrees of superheating	Superheated temperature	Saving of fuel compared with saturated steam	Remarks
1	2	3	4	5	6	7	8	9	10
Martin***)	1854	Hudson River	Steamer "Joseph Johnson"	1.3	124	49	173	33%	Saving exaggerated
Hirn*)	1856	Colmar	Stationary engine Woolf's system	3.00	143	102	235	27.5%	
Penn**)	1857	Greenwich	Steamer "Valetta"	1.4	125	56	181	20%	
P. & O. Co.***)	1860	Atlantic	Various Steamers	—	—	—	—	21–34%	Saving exaggerated

*) Zeitschrift des Vereines deutscher Ingenieure. 1866. No. IV.

**) SENNET. The marine steam engine. 1885. P. 196.

***) SCHWARZ-FLEMMING. Die Kesselabtheilung auf Dampfschiffen. 1873. P. 167.

These results about agree with the saving of 23 % which RANKINE*) calculates from his theoretical investigations for steam of 2.4 kilos per □cm superheated 80°. In general the saving of coal at sea *due to superheating alone* was calculated at 10 %, with trunk engines at only 5 % even.

21) Nevertheless the use of superheaters was valued so highly (chiefly on account of the reduction in the losses of steam mentioned in 16) that they were considered indispensable so long as the working pressure in marine boilers did not exceed 2 kilos per □cm (30 ℔s per □″). All high-class marine engines, whose cylinders had jackets were also provided with superheaters, although their first cost, as they were arranged in ships of war, was very considerable and their maintenance expensive, owing to the rapid stoppage of the tubes. **Former use of superheaters.**

22) As, soon after the use of superheaters became general, it was found to be impracticable to superheat low pressure steam more than 30° to 40° for any length of time, the necessity arose in the course of years, during which the working pressure and saturation temperature were rising, to gradually reduce this range of temperature available for superheating, in order to protect the engines from the bad consequences of using too highly superheated steam. But the more closely the saturation temperature approached the practicable superheated temperature, the smaller became the advantages of superheating. To this came the fact that as the working pressure rose the cut-off was made earlier than formerly; consequently the gain described in 15) had scarcely any more weight and that described in 16) became less and less appreciable. These circumstances, which gradually reduced the advantages of superheating to nothing, ultimately brought about the discontinuance of superheaters and at the present day no high-pressure boiler is fitted with one. Even steam driers are becoming rare, as it is desirable, for reasons to be later on explained, to have moist rather than too dry steam in the cylinder. **Later discontinuance of superheaters.**

23) In the same way, the recent attempts of NORMAND**) at Havre to superheat the receiver steam in compound and triple expansion engines must be described as a failure. NORMAND, in his engines, conducted the steam on its passage from one cylinder to the other through a superheater placed in the uptake. The complicated pipe-connections required for this **Experiments with superheated receiver steam in compound and triple expansion engines.**

*) J. W. M. RANKINE. A manual of the steam engine. 1873. P. 437.

**) M. DEMOULIN. Nouvelles machines marines des bâtiments à grand vitesse. 1887. P. 52.

prevented an extensive use of the arrangement and the perpetual repairs to the superheater tubes, combined with the slight economy obtained, led to the apparatus being taken out of most of the steamers fitted with it.

Arrangement of superheaters.

24) The arrangement of superheaters and their erection on board is described further on under Boiler Construction.

§ 14.

Mixtures of steam and air.

Where they occur.

1) In the condensers of steam engines there is always air mixed with steam and the experiment has recently been tried (with a view to economy of steam) of mixing as thoroughly as possible a certain quantity of compressed air with the steam before its admission into the cylinder.

The law for mixtures of this kind is therefore given here in a form corresponding to this special case.

Dalton's Law.

2) **Dalton's Law.** *A space of a certain magnitude is always capable of receiving the same quantity of steam whether (before the admission of the steam) it is exhausted of air, or contains air of any pressure whatever. The pressure p of the resulting mixture of steam and air is equal to the sum of the pressures which the steam on the one hand, and absolutely dry air on the other hand, would assume, if either of them alone occupied the space.*

$$p = p_d + p_l \quad \ldots \ldots \ldots \quad (67)$$

Condition of the steam in the mixture.

3) The steam in the mixture can be either saturated or superheated. The steam contained in atmospheric air is usually in the latter condition. Its temperature t is that of the air. The temperature t_0 at which is becomes suturated is called the *dew-point.*

Mixture with saturated steam.

4) *If the mixture contains saturated steam* then the pressure p_d of this steam and its density γ_d are at once known from its temperature t_0 by the steam tables on p. p. 28 to 31.

Mixture with superheated steam.

5) *If the mixture contains superheated steam*, we must first determine the dew-point t_0 of the latter in order to arrive at its density γ_u. The pressure p_d of the superheated steam follows from the temperature t_0 of its limiting state, and its specific volume for this state is then found by the equation of state for saturated steam (Eq. 56ª):

$$v_0 = \frac{R\left(T_0 - 37{,}79475\sqrt[4]{p_d}\right)}{p_d}$$

For superheated steam of the same pressure p_d and temperature t as the mixture, the specific volume is

$$v_u = \frac{R \cdot \left(T - 37{,}79475 \sqrt[4]{p_d}\right)}{p_d}$$

whence follows

$$\frac{v_\sigma}{v_u} = \frac{\gamma_u}{\gamma_d} = \frac{T_\sigma - 37{,}79475 \sqrt[4]{p_d}}{T - 37{,}79475 \sqrt[4]{p_d}}$$

$$\gamma_u = \gamma_d \frac{T_\sigma - 37{,}79475 \sqrt[4]{p_d}}{T - 37{,}79475 \sqrt[4]{p_d}} \quad \ldots\ldots\ldots\ldots (68)$$

6) *The pressure p_l of the air in the mixture* follows, by Eq. 67, after the pressure p_d of the saturated or superheated steam (in this case the same) has been determined, Pressure and weight of the air in the mixture.

$$p_l = p - p_d$$

and its density γ_l is then, by Eq. 17 page 13

$$\gamma_l = \frac{1}{v} = \frac{p_l}{RT} = \frac{p_l}{29{,}272\, T} \quad \ldots\ldots\ldots\ldots (69)$$

7) Such experience as we have hitherto of the behaviour of various mixtures of steam and air points to the conclusion that during admission less steam is condensed on the walls of the cylinder than is the case with saturated steam of the same pressure which is free from air. REYNOLDS *), who has made experiments on the influence of the aëration of steam upon its condensability on cool metallic surfaces, says that the condensation of a mixture of steam and air on such surfaces is hindered in consequence of a film of air being formed by molecular attraction on the metallic surface, and that this film of air being a bad conductor of heat must be penetrated by the steam which is to be condensed. WERNER **) imagines that the air is first deposited in the form of a net-work spread over the metallic surface. Experience with mixtures of steam and air.

8) The conclusions drawn by REYNOLDS from his experiments are as follows: Reynolds's Experiments.

a) A small admixture of air considerably reduces the condensability. This is only limited, in the case of pure steam, by the capacity of the cold surface for conducting heat.

*) London, Edinburgh and Dublin philosophical magazine for 1874.

**) Zeitschrift des Vereines deutscher Ingenieure. 1887. P. 285.

b) The condensability is reduced in about the same proportion as the steam pressure is increased by the air pressure, between the limits of 2 and 10%. From there the reduction is decreased more and more till an increase of pressure (of the steam by the air) of 30% is reached, at which point the condensability is at a minimum and remains so for any further increase.

c) By mixing air with the steam supplied to an engine the condensation of steam in the cylinder is considerably diminished and the efficiency of the engine increased.

d) The highest efficiency is attained when the pressure of the air in the mixture is about $^1/_{10}$ of the pressure of the steam or when about 0.125 cbmetres of air of atmospheric pressure at 16° C. are mixed with every kilo of steam, i. e. when $\frac{\text{weight of steam}}{\text{weight of air}} = 0.1525$.

Zeller and Hunt's Experiments.

9) Besides REYNOLDS'S experiments there are those of Engineers ZELLER*) and HUNT of the United States Navy with a horizontal noncondensing engine using steam at two to three atmospheres absolute pressure. These experiments gave no satisfactory results, the consumption of steam being no smaller with aërated than with pure steam. Besides, there was the expense of fitting and keeping up the air compressing pump. WERNER attributes this failure to the circumstance that the steam and air were not thoroughly mixed and that the proportion of the weight of air to that of steam was much too large; viz 0.37 and 0.88 instead of 0.1525, which REYNOLDS gives as the best.

Werner's suggestions.

10) WERNER is of opinion that in further experiments with aërated steam care should be taken that

a) the air should be distributed as evenly as possible throughout the steam in the proportion found to be most suitable and that the mixture introduced into the cylinder should always remain exactly the same;

b) that this object should if possible be attained without a special pump for compressing the air,

c) either by allowing the necessary quantity of air to be drawn through the feedpump, which must work continuously,

d) or by arranging the feedpump so that it will force the air (separated from the feedwater) either into the steam-

*) Journal of the Franklin-Institute. June 1886.

space of the boiler or the steam-chest of the engine in a stream consisting of finely divided jets.

11) With regard to these suggestions, I have to remark that in my opinion aërated steam is only adapted for noncondensing engines, because air, mixed with the steam in a condensing engine, besides impairing the vacuum in the condenser, would pass into the boiler with the feedwater, which is certainly to be avoided, as aërated water and moist air have been shewn by the unvarying experience of all the navies in the world to be the very worst destroyers of the internal parts of boilers. For this reason it is also advisable in the case of a noncondensing engine that the air be supplied by the feedpump separately from the water and introduced directly into the steam-chest of the engine. Applicability of aërated steam.

12) In conclusion it may be mentioned that HAYDEN-HANSBROUCK*) of New-York has taken out a patent for a steam-and-air engine into the steamchest of which he admits a small quantity of compressed air shortly before each end of the stroke and claims by this means to economise fuel. Steam-and-air engine.

*) Verhandlungen zur Beförderung des Gewerbfleisses. 1887. P. 79.

Third Division.

The Process in the Cylinder.

§ 15.

Influence of the cylinder walls.

Literature.

1) The pioneer experiments and investigations of CLARK*) and ISHERWOOD**) and still more those of HIRN and HALLAUER***) have brought out the fact, treated in extenso in GRASHOF'S****) newer works, that the behaviour of steam in the cylinder (or cylinders) of an engine depends in a very important degree upon the condition of the internal surfaces or walls of the cylinders, receivers, etc. in contact with it. According as the cylinder walls are cooler or hotter than the steam they enclose, i. e. according as they are jacketed or not, they give rise to phenomena, described somewhat as follows by BRAUER†) at the general meeting of the Verein Deutscher Ingenieure at Dortmund in 1883.

Cold cylinder walls.

2) **I. Cylinder walls cooler than the steam.** If the steam comes in contact with cylinder walls of a lower temperature than its own, then under any circumstances a part of the heat of the steam is transmitted to the walls. The rapidity of this transmission of heat is influenced

a) by the condition of the steam,
b) by the conductivity (for heat) of the walls,
c) by the "head" of temperature.

Condition of the steam.

3) a. *The condition of the steam* is so far of importance that superheated steam behaves like a permanent gas when the tempe-

*) D. K. CLARK: Railway machinery 1852.
**) B. F. ISHERWOOD: Experimental researches in steam engineering 1862.
***) Bulletin de la société industrielle de Mulhouse 1873—80.
****) Zeitschr. d. Ver. deutscher Ingenieure 1883 P. 161 & Zeitschr. d. Vereines deutscher Ingenieure 1884 PP. 263 & 313.
†) Zeitschr. d. Ver. deutscher Ingenieure 1883 P. 649.

rature of the walls is higher than the saturation temperature corresponding to the pressure of the steam, so that the transmission of heat takes place principally by radiation, because the layers of steam next to the cylinder walls soon assume the temperature of the latter and impede the further transmission of heat. But if the temperature of the walls is lower than the saturation temperature of the steam, then the layer of steam adjacent to the walls will be brought in a very short time to their temperature and condensed, whether the steam is originally saturated or superheated. The resulting film of water occupies only a very small portion of the space filled by the layer of steam (about a thousandth at 1.75 atmospheres pressure) from which it came and furnishes only an infinitesimal resistance to further transmission of heat, partly on account of its very small thickness, partly on account of the higher conductivity of water than of steam.

4) b. *If the conductivity of the walls* is very high, i. e. if the heat penetrates them rapidly, the succeeding layers of steam will condense as fast as the first. As the steam can follow up with a velocity which may be called infinite, the same, in fact which it would assume if flowing into a condenser of the same temperature as the walls, the rapidity of the whole process must be very great if the walls are good conductors. The steam is in this manner literally sucked up by the walls and very speedily deprived of its latent heat which goes over into the body of the cylinder. High conductivity of the walls.

5) If, on the other hand the conductivity of the wall is low, the heat in penetrating it will accumulate at the surface and soon restore there the saturation temperature of the steam, so that any more condensation on the surface of the wall only becomes possible when the further transmission of heat into the body of the wall has again reduced its surface temperature. Low conductivity of the wall.

6) If we imagine Fig. 1 Plate 2 to represent the section of a cylinder wall 30 mm thick, of the temperature t, suddenly exposed to a current of steam of temperature t_1, we shall have, by the laws of the propagation of heat, in a certain minute space of time a rise of temperature, which we may express by curve 1, in the body of the wall. This curve gradually goes over into 2, 3, 4, 5 and becomes at last the straight line 6, provided no heat is conducted away on the outside of the wall which will now have assumed the temperature 6 throughout. Curves illustrating the propagation of heat.

7) If, for instance, the temperature, at a distance x from the outside, is τ, and at distance $x + dx$, the temperature is "Head" of temperature.

$\tau + d\tau$, the temperature rises $d\tau$ for the length dx, so that $\frac{d\tau}{dx}$ is the "*head*". This latter therefore represents the tangent at any point of the temperature curves.

Variability of the head.

8) c. *To the head of temperature* the velocity is proportional at which the heat in the body of the cylinder wall is propagated or — *flows*. In the different curves this head diminishes from the inside to the outside, so that it is always greatest at the place of admission i. e. in the region of the steam ports. But there too the head, though infinitely great at the first moment, gets flatter and flatter until it ultimately, or strictly speaking, after an infinitely long time, becomes zero.

Heat-equilibrium.

9) If the cylinder wall is exposed on the outside to temperature t and can give up heat there as readily as it receives it on the inside, then the temperature curve will tend to approach the straight line O representing *heat-equilibrium*. When this equilibrium is established, the "head" is the same at every point, so that the heat imparted at any one point equals the heat withdrawn and the temperature remains unaltered.

Amount of heat transmitted.

10) Assuming for instance that the difference between the internal and external temperatures, $t_1 - t = 30^0$, we get, per running millimeter of the wall 30 mm thick, a "head" of $\frac{30}{30} = 1 = tan.\ 45^0$. Experiments have shewn that in this case about 10 thermal units (according to KIRSCH*) 16) will pass into the wall per sq. meter of steam-covered surface per second. But as with the steam suddenly acting, curves of form 1 are at first produced and the "head" at admission may easily be 10 to 100 times greater than unity, as many as 100 to 1000 thermal units per second may be absorbed by the iron wall if only the conductivity of the latter were to be considered.

Hot cylinder walls.

11) **II. Cylinder walls hotter than the steam.** If steam enters a cylinder whose walls are hotter than the steam, heat will be imparted to it from the hot walls. The rapidity of this transmission of heat depends principally upon whether the wall is

d) perfectly dry, or

e) covered with a film of condensed water.

Dry cylinder walls.

12) d. *If the wall is perfectly dry* the transmission of heat from it to the steam can only take place by the slow process of

*) KIRSCH. Die Bewegung der Wärme in den Cylinderwandungen der Dampfmaschine. Leipzig 1886. P. 7.

radiation and conduction through the mass of the steam, and is therefore infinitesimal.

13) e. *If the wall is covered with a film of condensed water*, the water will have approximately the same temperature as the wall and will therefore also be hotter than the enclosed steam. As the latter, in accordance with its lower temperature, also has a lower pressure than it would have if the temperature of the walls were its own temperature of saturation, it follows that the film of water on the walls, being under this low pressure, must boil. The evaporation of the film of water now proceeds at the expense in the first place of its own heat, afterwards with the assistance of the heat of the wall which is rapidly furnished as it is taken up. So long as a breath of moisture is present, an intense emission of heat from the walls is possible, but from the moment they are dry, only the unimportant radiation remains. **Wet cylinder walls.**

14) If a larger quantity of water is in the cylinder than can support itself on the walls by adhesion, it collects at the lowest portions of the cylinder and its upper surface is in contact with the entering steam. But the state of equilibrium is not disturbed by the water being heated from *above*. It therefore remains pretty free from the circulation which would promote transmission of heat, even when it is agitated more or less by the piston touching it. As the thermal conductivity of water at rest is only $^1/_{20}$ to $^1/_{30}$ of that of iron, there is scarcely any doubt that *the presence of a considerable quantity of water in the cylinder rather retards than accelerates the transmission of heat.* If the water remains for a length of time in the cylinder, it can only evolve steam to any great extent, so long as the temperature of the water is higher than that of the steam (admitted to the cylinder). As, however, the water only takes up slowly the heat of the steam above it, and on boiling easily parts with it again, we may assume that the temperature of any such water (at the bottom of the cylinder) will be a little above the lowest temperature in the cylinder i. e. that corresponding to the back-pressure. **Water in the cylinder.**

15) **III. Periodic motion of heat in the cylinder-walls.** The water-value (see § 5, 5) of a steam-cylinder is, in consequence of its great weight, so considerable that, with an early cut-off, the total heat of the steam admitted per stroke would hardly suffice to raise the temperature of the cylinder much more than 1°. On the other hand, the temperature of the internal cylinder wall oscillates at every stroke with the temperature of the **Oscillations of temperature in the cylinder walls.**

steam, i. e. between its highest and lowest temperature corresponding to the admission and exhaust, so that the wall may be subjected to variations of from 50° to 100°, according to the initial and terminal pressures. Considering the above-mentioned large water-value of the cylinder, these oscillations of temperature can only extend to a comparatively slight depth into the wall, the external layers of which, beyond this range, must assume a mean temperature depending upon the outside conduction of heat.

Region of the oscillations of temperature.

16) The depth of the internal layer exposed to the influence of the oscillations of temperature taking place on the surface, — or the depth to which the periodic movement of heat extends — depends upon the rapidity with which the periods — or strokes — succeed each other. This depth, according to KIRSCH*) is inversely proportional to the square root of the number of the revolutions. He calculates that for the following revolutions and thicknesses, it is this fraction, viz.

	0,01	0,001	0,0001 of the thickness of the wall	
for	19 mm	29 mm	38 mm and	20 revolutions per minute.
"	12 "	18 "	24 " "	50 " " "
"	9 "	13 "	17 " "	100 " " "
"	6 "	9 "	12 " "	200 " " "
"	4 "	6 "	8 " "	500 " " "

On comparing these values with the usual thicknesses of cylinders, we may safely assume that *the thickness of the wall does not influence the periodic portion of the movement of heat.*

Accumulation of heat in the cylinder wall.

17) If, in a cylinder, a certain quantity of heat is given up to the cylinder wall by condensation of steam, this quantity of heat can only be exactly equal to that quantity necessary for the re-evaporation of the water resulting from the above condensation in the case of the conditions of pressure, under which the condensation and re-evaporation take place, being identical. But in a steam cylinder, for reasons which will be shewn immediately, the condensation always takes place under a higher pressure than the re-evaporation, wherefore the wall receives more heat than it gives off, in consequence of the greater total heat of the higher-pressed steam (during admission than during expansion and exhaust) and thus should gradually become hotter. The difference between these two heat-quantities is nevertheless so slight, that even a carefully

*) KIRSCH. Die Bewegung der Wärme in den Cylinderwandungen der Dampfmaschinen. Leipzig 1886. P. 10.

cleaded cylinder probably loses more heat on the outside by conduction or radiation than it receives from excess of heat of condensation. In a jacketed, or otherwise heated, cylinder this heat cannot escape on the outside, but is used in evaporating the water deposited on the cylinder wall (without giving up heat), which arises either from the moisture of the steam or is precipitated from it during expansion. Under normal circumstances therefore, the precipitation of water on the internal cylinder wall will only be just so great as to cause the heat given *up* to the cylinder wall by its condensation to be again given *out* for its re-evaporation, so that neither an accumulation of heat *in* the cylinder wall nor of water *on* the surface of it, is possible.

18) **IV. Movement of heat in the walls of unjacketed cylinders.** Let Fig. 2 Plate 2 represent, according to GRASHOF *), the section of a cylinder wall at, or near, one end. On the outside, at B it is rendered non-conducting by cleading, and therefore assumes from B to C a constant temperature t (between the admission temperature t_1 and the exhaust temperature t_2), which must be regarded as the mean temperature for the layers between C and A exposed to the periodic movement of heat. Conditions of temperature.

19) The steam, on entering the cylinder from the boiler always comes into contact with wall-surfaces which were just before in connection with the condenser or the air and therefore possess a lower temperature than the steam. If the curve 4 represents the distribution of temperature in the cylinder wall at the end of the exhaust, it is clear from the foregoing that the steam must experience a partial condensation during admission, in consequence of which a film of water forms on the internal surface. Its heat of condensation raises the temperature of the innermost layer of the wall suddenly to about the temperature t_1 and by gradual penetration into the wall causes the curve 4 to go over into the curve 1 at the moment of cut-off. If now at the beginning of the expansion the temperature of the steam, and therefore also that of the innermost layer of the wall, sinks, then with the great head of temperature shewn by Curve 1, the movement of heat in the cylinder wall will go on towards the outside until it changes the curve of distribution of temperature 1 into curve 2. This transformation occupies the greater part of the period of expansion, so that a transmission of heat from the Movement of heat during admission and beginning of expansion.

*) F. GRASHOF. Theoretische Maschinenlehre. Hamburg and Leipzig 1888. Vol. III. P. 551.

wall to the film of condensed water can only begin to take place towards the end of this period.

Movement of heat during the remainder of the expansion and the exhaust.

20) The intense transmission of heat from the wall to the water causes the latter to evaporate again according to the fall of pressure during the expansion. *Re-evaporation* begins and changes curve 2 into curve 3, which serves to illustrate the distribution of temperature in the wall at the close of the expansion or beginning of the exhaust. As in fast-running engines with negative inside lap, like marine engines, the exhaust begins at 80 to 95 % of the stroke, re-evaporation in their case generally takes place during exhaust or the return stroke of the piston, with the back pressure in the cylinder (of an ordinary condensing engine) from 1.25 to 2.5 ℔s per □" absolute, and the watery particles on the cylinder walls evaporating at 50° to 70° C. By the re-evaporation all the heat is again withdrawn from the wall which it had taken up from the steam during admission, so that at the close of the exhaust the curve of distribution of temperature 3 must have again assumed the form 4. Therefore, of the whole heat quantity periodically exchanged between the steam and the cylinder wall, which may be represented by the area contained between curve 4 and curve 2, only the small portion bounded by curves 2 and 3 is converted into energy during the expansion, while the remaining greater part is lost without doing any work at all. But the conversion into useful work of even the small quantity of heat gained during expansion involves a sacrifice, because the re-evaporation of the film of water takes place under a lower temperature than that at which it was condensed, so that a corresponding useful "head" of temperature is lost.

Chief cause of the losses of steam.

21) As all recent investigations have shewn, the initial heavy condensation of steam and the subsequent re-evaporation of the water resulting from it are the cause of the considerable loss of steam existing more or less in every engine, compared with which the losses arising from leaky slides and pistons are very trifling in well made engines. This internal condensation is largely increased by the circumstance that the surface of the cylinder, after being dried by the re-evaporation, is made clean and bright by the friction of the piston, so that it is again rendered highly effective for condensing the entering steam.

Conditions of temperature.

22) **V. Movement of heat in the walls of jacketed cylinders.** If the jacket is filled, as in most cases, with boiler steam, the tem-

perature of the external layer of the cylinder wall at *B*, Fig. 3, Plate 2, is equal to that of the entering steam t_1 and in consequence of the continuous movement of heat, the curve of distribution of temperature takes a falling direction from *B* through *C* to *A*. The portion *BC* of the curve shews the constant temperature of the external layers, while *CA* may be taken to represent the mean temperature of the internal layers. The point *C* shews the depth to which the periodical movement of heat in the internal layers extends, and this depth is less than in the unjacketed cylinder (Fig. 2).

23) In a jacketed cylinder also, the innermost layer of the wall must be brought at the beginning of the admission to the admission temperature t_1 by condensation. But on account of the higher average temperature of the wall (Fig. 3) the curve of distribution of temperature 1 which refers to this moment (admission), is less steep than in Fig. 2. The lower head of temperature in the innermost layers of the wall allows of a quicker reversal of the heat-movement from outwards to inwards, which takes place soon after the beginning of the expansion and transforms curve 1 into curve 2. The heat given up to the cylinder wall during the expansion, represented by the area $\alpha\beta\gamma$ bounded by curves 1 and 2, is therefore much smaller than in the unjacketed cylinder (Fig. 2). As the expansion goes on, the heat is again given up by the wall and evaporates the precipitated water, so that a considerable useful effect is produced by the re-evaporation. At the opening of the exhaust, just before the end of the stroke, the curve of distribution of temperature 2 is changed to 3. During the exhaust the last residue of water on the wall very quickly evaporates, whereupon curve 3 takes the form 4. By this time the whole of the heat given up to the wall by the steam during admission has again left the wall, and now only that heat requires to be conducted off which passes during one revolution of the engine from the jacket through the body of the cylinder to its inner surface. But as the inner surface has now become dry, this transmission of heat to the (exhausting) steam can only take place slowly, therefore an accumulation of heat takes place in the internal layers and by the time the exhaust closes curve 4 is transformed into curve 5. **Exchange of heat during the different portions of the stroke.**

24) The quantity of heat which escapes from the internal wall-surface during a revolution is represented in Fig. 3 as the area bounded by the curves 2 and 5. That portion of the heat which corresponds to the area between curves 2 and 3 is **Magnitude of the heat quantities exchanged.**

again made useful in the re-evaporation during expansion, whereas the area between curves 3 and 5 shews the heat lost during the exhaust ("the exhaust-waste"). The portion of this heat which comes through from the jacket is expressed by the area between curves 4 and 5. A comparison between Figs. 2 and 3 shews that the area representing the heat recovered during the expansion is greater in the jacketed than the unjacketed cylinder, and that on the other hand, the "exhaust-waste" area is much smaller with the jacketed than the unjacketed cylinder.

Advantage of the jacket.

25) *The advantage of the jacket heated with boiler steam*, as we see from the foregoing, *is, that in consequence of the higher mean temperature of the internal surface of the cylinder, the condensation during admission is diminished and most of the resulting film of water evaporated during expansion, thus producing useful work.* If BRAUER'S*) view is correct, viz that this film of water, in a jacketed cylinder, follows the piston like a shadow, disappearing as soon as it is produced, — the capacity of the wall for parting with its heat must be considerably curtailed, as without the watery film it can only lose trifling quantities of heat. It is therefore clear that the losses of heat, and therefore also of steam, in a jacketed cylinder are much smaller than in an unjacketed one, which is corroborated by all the experiments described in § 16.

Heat movement in the middle portion of the cylinder.

26) **VI. Mean temperature of the wall.** As was expressly premised in 18), the movement of heat explained in the foregoing only takes place in the portions of the cylinder wall adjacent to the cover or the bottom, or forming annular elements of the wall in the neighbourhood of the ends. For annular elements of the wall situated at about half stroke there is a more frequent reversal of the heat movement according to the particular position of these elements.

Possible cases.

27) If $H =$ the whole stroke,

$h =$ the portion of the stroke completed at cut-off, and

$x =$ the distance of an annular element from the cylinder cover,

the following cases are possible.

a) The piston is on the down stroke and $x > h$, then the annular element is first in contact with the exhausting and later with the expanding steam. But if $x < h$ the

*) Zeitschrift des Vereines deutscher Ingenieure 1883, P. 657.

element is first in contact with exhausting, then with entering, and finally with expanding steam.

b) In the up stroke, for $H - x > h$, we have the first case of a) and for $H - x < h$, the second case of a).

c) Finally, if $h < 0.5\,H$ and $h < x$ as well as $x < H - h$, the element will never come in contact with the entering steam at all.

28) From this it follows that an element will in general meet the cooler steam, the closer it is situated to the middle of the stroke. *Therefore the temperature of the wall must increase from the middle towards the ends, and the more so the earlier the cut-off is.* As a rule this difference of temperature only amounts to a few degrees. There is thus, besides the periodic heat-movement in a radial direction from outwards to inwards, or vice versâ, simultaneously a lesser longitudinal heat-movement in the wall, chiefly directed from the ends towards the middle. **Longitudinal heat-movement.**

29) In the absence so far of a formula for the mean wall temperature founded on theory, WERNER*) has propounded the following empirical formula which is said to agree well with experience. The cylinder is assumed to be completely enveloped in some badly conducting substance. The formula is based on the supposition that the end of 17) is true, viz that the whole of the heat taken up by the wall during admission or later, is again parted with during expansion and exhaust. **Formula for the mean temperature of the wall.**

30) Let **Definitions.**

t_m = the mean wall temperature of an unjacketed cylinder,
t_{m_1} = " " " " " a jacketed " ,
t_1 = " admission temperature of the steam,
t_2 = " exhaust " " " " ,
φ = in circular measure, the angle of the crank from the dead-point, at cut-off,
π = the number 3.1415,
$\varkappa$ = a coefficient, explained below.

31) *For an unjacketed cylinder* we have then **Formula.**

$$(t_1 - t_m)\,\varphi = (t_m - t_2)\,\pi$$

and thence

$$t_m = t_1 - (t_1 - t_2)\frac{\pi}{\varphi + \pi} \quad \ldots\ldots\ldots \quad (70^a)$$

*) Zeitschrift des Vereines deutscher Ingenieure 1883. P. 267.

For a jacketed cylinder, heated with boiler steam

$$t_{m_1} = \frac{t_m + x\, t_1}{1 + x}$$

or

$$t_{m_1} = t_1 - \left(\frac{t_1 - t_2}{1 + x}\right) \frac{\pi}{\varphi + \pi} \quad \ldots\ldots\ldots \quad (70^b)$$

The coefficient x

32) The coefficient x approaches unity the more closely,

a) the more completely the jacketing is effected. To complete jacketing belongs not only the heating of the outside of the cylinder (by *passing* boiler steam *through* the jacket and *not merely admitting steam to it*), but also the heating of the cylinder ends and piston;

b) the earlier the compression begins;

c) the more highly the steam is superheated at admission, — if superheating is resorted to.

The arc φ

33) *The arc* φ, if $\varepsilon = \frac{h}{H}$ the given cut-off, and the connecting rod is regarded as infinitely long (which is quite sufficiently exact for the value of t_m, itself only an approximation), is found thus

$$h = \frac{H}{2} - \frac{H}{2} \cos\varphi = \frac{H}{2}(1 - \cos\varphi)$$

$$\cos\varphi = 1 - \frac{2h}{H} = 1 - 2\varepsilon \quad \ldots\ldots\ldots\ldots \quad (71)$$

§ 16.

Value of jacketing.

Example.

1) **I. Losses of steam in unjacketed cylinders.** The mean temperature of the cylinder wall of an unjacketed low-pressure engine is calculated as follows by Eq. 70a. Let the working pressure of the boiler be 2 kilos per □ cm (say 30 ℔s per □″), and accordingly the admission pressure in the cylinder 1.5 k per □ cm or 2.5 k absolute, so that $t_1 = 127^0$. Let the exhaust temperature t_2 be 60^0, corresponding to a condenser back pressure of 0.2 k. Let the cut-off, most favourable for such an engine, be $\varepsilon = 0.35$. Then, by Eq. 71:

$$\cos\varphi = 1 - 2 \,.\, 0{,}35 = 0{,}3 = \cos 73{,}5^0$$

and therefore

$$\varphi = \frac{73{,}5 \,.\, \pi}{180} = 1{,}2828.$$

From this we get

$$t_m = 127 - (127 - 60)\frac{3{,}1415}{1{,}2828 + 3{,}1415}$$

$$t_m = 127 - 67\frac{3{,}1415}{4{,}4243} = 127 - 47{,}5$$

$$t_m = 79{,}5^0$$

2) It is thus seen that in such an engine, the steam, entering the cylinder at 127^0 comes in contact with walls $127^0 - 79.5^0 = 47.5^0$ cooler than itself. Consequently it parts with a very considerable quantity of heat to the walls, as described in § 15, 19, and a large amount of condensation during the whole period of admission and the beginning of the expansion takes place, followed by much injurious re-evaporation during the exhaust. The various heat quantities exchanged during this process between the steam and the wall, may be determined by the method explained in the next paragraph. For the present, it is sufficient to have proved that the exchange of heat between the steam and the wall means losses of heat and therefore losses of steam. **Cause of the losses of steam.**

3) The losses of steam thus arising in unjacketed cylinders have been determined by the experiments on the Sloop "Michigan" of Engineer ISHERWOOD of the U. S. Navy. These experiments were carried out at a piston speed of 1.14 m per sec. with saturated, or rather partially moist steam and gave results shewn in the table below. **Isherwood's Experiments.**

Table of the losses of steam in unjacketed cylinders.

Cut-off	0,93	0,64	0,40	0,354	0,25
Ratio of expansion	1,07	1,50	2,50	2,85	4,0
Water used per stroke, by direct measurement in ℔s	5,20	4,40	3,40	3,63	3,02
„ „ „ „ „ „ in kilos	2,36	1,99	1,54	1,64	1,37
„ „ „ „ taken from the indicator cards in ℔s	4,42	3,15	1,84	1,62	1,17
„ „ „ „ „ „ „ „ cards in kilos	2,00	1,43	0,83	0,74	0,53
Loss of steam per stroke, by condensation, leaks &c. in ℔s	0,78	1,25	1,56	2,01	1,85
„ „ „ „ „ „ „ „ „ in kilos	0,35	0,56	0,71	0,91	0,84
Percentage of loss of steam to actual steam used	15	28	46	55	61

4) Although these losses were rather high, owing to the low piston speed and the moisture of the steam, the table nevertheless shews that with an increasing ratio of expansion the **Inferences from these experiments.**

loss of steam may rise to such an extent in consequence of the augmented difference of temperature between the entering steam and the wall, that for a cut-off below 0.4 the loss is actually greater than the useful steam consumption.

Experiments of Gately and Kletzsch.

5) Some more very thorough investigations into the extent of steam-losses in an unjacketed cylinder were made by GATELY and KLETZSCH*) under the direction of Prof. THURSTON in May 1885 with a jet-condensing single-cylinder Harris-Corliss engine at Sandy-Hook. They shewed that the loss x expressed as a percentage of the total steam used might be computed, within the limits of the experiments with this engine by means of the following empirical formula:

a) *for variable cut-off* $\varepsilon = 0.15$ to 0.55, working pressure about 4 kilos per □ cm (say 60 lbs), and Revolutions 68,

$$x = 19\sqrt{\frac{1}{\varepsilon}}\ \% \quad ; \ldots\ldots\ldots\ldots (72)$$

b) *for variable boiler-pressure* $p = 1.57$ to 5.62 k per □ cm, cut-off ranging from 0.24 to 0.20, and revolutions about 70

$$x = 45 - 1{,}8\,p\ \% \quad ; \ldots\ldots\ldots\ldots (73)$$

c) *for variable number of revolutions per min.* $N = 34$ to 63, cut-off ranging from 0.93 to 0.98, and working pressure about 1.33 k per □ cm,

$$x = 45 - 0{,}33\,N\,\% \ldots\ldots\ldots\ldots (74)$$

Inferences from these experiments.

6) The experiments confirm the well-known fact that, other things being equal, the loss of steam in an unjacketed cylinder increases in general, as

a) the cut-off is made earlier with the same pressure and piston speed;

b) the piston-speed is reduced with the same pressure and cut-off.

According to Eq. 73 the loss of steam in the unjacketed cylinder of the engine experimented upon increased as the pressure was reduced with the same cut-off and piston-speed, an observation opposed to all previous experience. *Ceteris paribus*, the loss of steam must increase with increasing pressure, and therefore temperature of admission, as is also proved by the more recent experiments of ENGLISH (see § 17, 34). A close examination of the results of the five experiments from which Eq. 73 is derived, shews too that there was a loss of steam of 48 % at 4.7 kilos per □ cm pressure, and 41 % at 2.6 kilos

*) The Engineer. 1885. II. PP. 341, 362, 402, 424, 492. 1886. I. P. 84.

pressure — a simple confirmation of the general rule. In developing Eq. 73 the result of the first named experiment is unfortunately put aside and the observed values from the other two experiments (less correct, in my opinion) are substituted for it and combined with the last named results.

7) But what these experiments chiefly bring out is that, other things being equal, the loss is inversely proportional to the square-root of the ratio of cut-off. The constant which, multiplied by $\sqrt{\frac{1}{\varepsilon}}$, gives the loss as a percentage of the total steam consumed (in this case 0.19), ranges, according to THURSTON's investigations, between the limits of 0.1 and 0.2 or thereabouts. For an unjacketed condensing engine, as in the present case, the value of the constant approaches the latter number. These data correspond very well with ISHERWOOD's results on the "Michigan", if the very moist steam and low piston speed he used are taken into account. Agreement of these experiments with those of Thurston and Isherwood.

8) KIRSCH (see § 15, 16) shewed by theory, and ENGLISH by practical experiments (see § 17, 30) also, that the loss is inversely proportional to the square-root of the number of revolutions — other things being the same. So that a universal law might be thus stated, — Law of the loss of steam.

"*Other things being equal, the loss of steam in unjacketed cylinders is inversely proportional to the square root of the ratio of cut-off and the square root of the revolutions.*"

9) As the tables on pp. 86 and 87 shew, and as may also be inferred from the Düsseldorf*) investigations with an unjacketed condensing engine, *it is bad practice to construct single cylinder expansion condensing engines without jackets* on account of the great losses of steam thus caused, which, it has been proved, may exceed 40 % of the total steam used. Single cylinder expansion condensing engines must have jackets.

10) **II. Losses of steam in jacketed cylinders.** The steam jacket was invented in the year 1769 by WATT, but not carried out in practice by him before 1776, from that time however, he retained it in all his engines. It nevertheless appears more than questionable whether WATT knew the necessity and action of the jacket so thoroughly as we do to-day; it is more probable that he only introduced it in order to carry out to the uttermost his intention of "keeping the steam in the cylinder as warm as possible". Introduction of the steam jacket by Watt.

*) Untersuchungen von Dampfmaschinen etc. der Gewerbe-Ausstellung in Düsseldorf 1880. Aachen 1881. P. 9.

Necessity of the jacket.

11) WATT's successors did not understand the advantages of the jacket, and as the construction of it with their limited appliances often caused difficulties, they abandoned it as superfluous. Not until attempts were made to utilize in a higher measure the expansive power of steam did the necessity of jacketing condensing engines become more evident. It then began to be generally introduced, so that at the present time most of the single cylinder expansion engines afloat are fitted with jackets.

Methods of jacketing.

12) The earliest jackets only surrounded the sides of the cylinder, because these, being always kept polished by the piston as mentioned in § 15, 21, were particularly suited to condense the entering steam and cause loss. Soon, however, the ends of the cylinder were made hollow and their internal spaces connected with the jacket. Later still, arrangements were devised for conveying the jacket steam into the hollow body of the piston, but, on account of their complication, did not find favour in practice. The construction of jacketed cylinders will be described later under cylinders.

Methods of heating the jacket.

13) The losses arising in jacketed cylinders depend upon the character of the vapours or gases employed in the jacket and also upon the method in which they are applied, i. e. whether the gases *pass through* the jacket in a current or *remain at rest in it.* The losses are smaller, other things being equal, when the heating vapours or gases pass through the jacket than when they remain in it at rest, because in the first case the gas or vapour robbed of its heat by the jacket walls is at once carried off and replaced by fresh vapour of the original heating power, whereby the cylinder is more effectively kept warm than in the second case.

Various heating agents employed.

14) *For heating cylinders* there have been used
- a) Furnace-gases,
- b) Exhaust steam,
- c) Steam taken from the steam chest of the engine,
- d) Steam taken direct from the boiler, and
- e) Steam hotter than that in the boiler.

Furnace-gases.

15) a. **The heating of the cylinder with furnace-gases,** apart from the difficulties connected with it, has been proved by experiment to be less effective than heating by steam. The transmission of heat from the (permanent) furnace-gases to the cylinder wall when they are at rest, takes place, by § 15, 3, chiefly through radiation, because the layers of gas next to the cylinder soon assume its temperature and obstruct any further passage of

heat. But even when the furnace-gases are made to traverse the jacket-space at a very high temperature, they are only of slight effect in raising the mean temperature of the cylinder wall, because their power of parting with heat is but small, in fact very much smaller than that of steam, as the following calculation shews; besides, it is known that the mean temperature of the cylinder wall is far less dependent upon the external temperature than upon the intensity (or rapidity) of the external transmission of heat. For whereas 1 cbm of steam at 152^0 temperature and 5 k per □ cm pressure weighs 2.6 k and, with its heat of evaporation of about 500 thermal units could give up $2.6 \times 500 = 1300$ thermal units to the cylinder wall, before being converted into water of 152^0, — 1 cbm of furnace gas of 350^0 will weigh about 0.66 k and (at a specific heat of 0.24) in cooling down to 150^0 will only give out $0.66 \times 0.24\,(350 - 152) = 31$ thermal units to the wall.

16) **b. The heating of the cylinder with exhaust steam** is said to have been adopted in former years in a few engines of the United States Navy. These were low-pressure engines, working at 1.33 to 1.5 k per □ cm (say 22 ℔s), so that the difference of temperature between the steam at admission and that exhausting through the jacket into the condenser was about 50^0 to 60^0. As in this case it is evident that heat must pass *from* the admission steam through the cylinder wall *into* the steam in the jacket, this arrangement turned out to be positively injurious. Certainly the loss of steam would have been less if this jacket had been omitted altogether and the cylinder had simply been well cleaded. **Exhaust steam.**

17) **c. Warming the cylinder with steam-chest steam** has been resorted to in many engines of the French Navy, those of the Ironclad Corvette "Alma", for instance, built about twenty years ago at the Chantiers et Ateliers de l'Océan at Havre. The steam is taken into the jacket and afterwards into the steamchest, an arrangement which under ordinary circumstances cannot be approved on account of the loss of temperature, and therefore pressure, which the entering steam suffers through parting with some of its heat in the jacket. This method of heating the cylinder can only be defended in cases where the steam cannot be passed through the jacket in any other way, or where the objectionable consequences of the use of highly superheated steam are thus guarded against in the cylinder by first cooling it in the jacket. — But the efficiency of the jacket is diminished by the use of superheated steam because such steam cannot give up as much heat to the moist walls **Steam chest steam.** **Superheated steam.**

(Continued on P. 88.)

Table of the arrangement and dimensions of the 5 American marine engines.

No.	Name of Ship	Date of Experiment	Engines				Cylinders		
			Age	System	Type	Jacket	No.	Diar. cm	Stroke cm
1	2	3	4	5	6	7	8	9	10
1	**Dallas**	August 1874	new	Single cylinder low-pressure	Inverted direct acting surface condensing	unjacketed	1	91,4	76,2
2	**Dexter**	August 1874	"	Single cylinder high-pressure	"	"	1	66,0	91,4
3	**Gallatin**	Januar 1875	4 years	"	"	sides, cover, & bottom jacketed	1	86,6	76,2
4	**Bache**	May 1874	4 years	Woolf's	"	*L. P.* cylinder completely jacketed *H. P.* cylinder not jacketed	2	40,6 high-pr. 63,5 low-pr.	61,0
5	**Rush**	August 1874	new	Compound	"	Both cylinders completely jacketed	2	61,0 high-pr. 96,5 low-pr.	68,6

Table*) of the steam losses in the single cylinder unjacketed engines of the steamers "Dallas" & "Dexter".

Drawing of the machins	Steamer "Dallas" at Low-Pressure			Steamer "Dexter" at Medium-Pressure			Steamer "Dexter" at High-Pressure			
1	2	3	4	5	6	7	8	9	10	11
Duration of each experiment, hours	1,52	1,55	1,60	1,32	1,20	0,92	2,92	1,42	34,50	0,65
W. P. of boiler in ℔s per □" engl.	35,40	35,29	33,70	40,62	39,90	41,87	68,70	69,29	67,12	66,42
„ „ „ „ k „ □ cm	2,49	2,48	2,37	2,85	2,80	2,94	4,83	4,87	4,71	4,64
Ratio of cut-off	0,133	0,197	0,288	0,262	0,381	0,452	0,183	0,233	0,248	0,333
„ „ expansion including clearance	5,067	3,894	2,936	3,338	2,423	2,084	4,457	3,669	3,489	2,724
Piston speed in feet per min. engl.	243,50	284,50	322,50	304,80	331,20	364,20	339,00	385,80	366,00	436,80
„ „ metres „ second	1,24	1,44	1,64	1,55	1,68	1,85	1,72	1,96	1,86	2,22
IHP	137,96	186,80	242,80	124,27	161,85	196,19	185,87	228,07	218,97	292,37
Steam used per IHP per hour by direct measurement in ℔s engl.	26,69	26,96	28,90	28,80	28,94	31,79	23,86	24,12	23,90	24,31
„ „ „ „ „ „ „ „ in kilos	12,11	12,23	13,11	13,06	13,12	14,42	10,82	10,94	10,84	11,02
„ „ „ „ „ „ cards in ℔s engl.	19,20	20,72	21,20	18,85	18,66	20,33	16,23	17,24	16,33	18,54
„ „ „ „ „ „ „ in kilos	8,71	9,39	9,62	8,55	8,46	9,22	7,36	7,82	7,41	8,41
„ lost „ „ „ in ℔s engl.	7,49	6,24	7,70	9,95	10,28	11,46	7,63	6,88	7,57	5,77
„ „ „ „ „ in kilos	3,40	2,83	3,49	4,51	4,66	5,20	3,46	3,12	3,43	2,62
Percentage of steam lost to total steam used as measured	28,00	23,20	26,64	34,55	35,57	36,00	32,00	28,52	31,70	23,74
Consumption of coal per IHP per hour in ℔s engl.	—	3,42	—	—	—	—	—	—	3,13	—
„ „ „ „ „ „ in kilos	—	1,55	—	—	—	—	—	—	1,42	—

*) Engineering 1875. I. P. 129.

Table*) of the steam losses in the single cylinder engine of the steamer "Gallatin", working with and without the jacket.

Average duration of each experiment two hours	Low-Pressure without jacket	Low-Pressure with jacket	Medium-Pressure without jacket	Medium-Pressure with jacket	Medium-Pressure without jacket	Medium-Pressure with jacket	High-Pressure without jacket	High-Pressure with jacket	High-Pressure without jacket	High-Pressure with jacket
1	2	3	4	5	6	7	8	9	10	11
W. P. of boiler in lbs per □" engl.	14,60	15,40	40,90	42,80	37,70	37,40	71,50	71,60	61,10	67,20
„ „ „ „ k „ □ cm	1,02	1,22	2,87	3,00	2,65	2,63	5,02	5,03	4,29	4,72
Ratio of cut-off	0,46	0,47	0,14	0,14	0,41	0,42	0,07	0,08	0,18	0,19
„ „ expansion including clearance	2,01	2,00	5,21	5,21	2,23	2,21	7,78	7,31	4,25	4,19
Piston speed in feet per min. engl.	200,10	206,30	231,00	230,10	278,70	291,10	261,60	255,50	306,80	343,50
„ „ „ metres per second	1,017	1,048	1,123	1,169	1,416	1,479	1,329	1,298	1,508	1,745
IHP	86,980	95,250	127,196	137,501	236,457	255,005	185,090	196,988	282,929	347,985
Steam used per *IHP* per hour by direct measurement in lbs engl.	40,37	33,34	26,70	24,02	28,10	26,52	25,03	20,49	23,99	21,48
„ „ „ „ „ „ „ „ „ in kilos	18,31	15,12	12,11	10.89	12.74	12,03	11,35	9,29	10,88	9,74
„ „ „ „ „ „ „ the cards in lbs engl.	25,62	25,97	18,49	17,23	22,21	21,43	17,50	15,67	16,95	17,18
„ „ „ „ „ „ „ „ „ in kilos	11,62	11,78	8,38	7,81	10,07	9,72	7,94	7,11	7,69	7,79
„ lost „ „ „ „ in lbs engl.	14,75	7,37	8,21	6,79	5,89	5,09	7,53	4,82	7,04	4,30
„ „ „ „ „ „ in kilos	7,14	3,34	3,72	3,08	2,67	2,31	3,41	2,18	3,35	1,94
Percentage of steam lost to total steam used as measured	36,54	22,10	30,74	28,26	20,96	19,19	30,08	23,52	29,34	20,02
„ „ „ „ „ „ „ „ in lbs engl.	4,66	3,85	3,12	2,81	3,28	3,09	2,95	2,41	2.82	2,53
„ „ „ „ „ „ „ „ in kilos	2,11	1,74	1,41	1,27	1,48	1,40	1,34	1,09	1,28	1,15

*) Engineering 1876. I. P. 121.

Table*) of the steam losses in the engine of the steamer "Bache" working as a two cylinder single expansion engine and as a Woolf's engine.

Average duration of each experiment two hours	Single-expansion						Woolf					
	without jacket	with jacket	without jacket	with jacket	without jacket	with jacket	without jacket	with jacket	without jacket	with jacket	without jacket	with jacket
1	2	3	4	5	6	7	8	9	10	11	12	13
W. P. of boiler in ℔s per □" engl.	78,11	79,50	79,62	81,10	81,00	80,80	80,31	80,12	80,28	80,33	82,00	81,37
„ „ „ in kilos per □ cm	5,49	5,59	5,60	5,70	5,69	5,68	5,64	5,69	5,64	5,65	5,76	5,72
Gross ratio of cut-off	0,155	0,163	0,096	0,081	0,0475	0,042	0,177	0,174	0,150	0,109	0,109	0,059
„ „ „ expansion, i. e. including clearance	5,32	5,11	7,62	8,57	11,82	12,62	5,634	5,73	6,658	9,20	9,146	16,85
Piston speed in feet per min. engl.	188,2	214,9	179,5	184,3	149,0	159,4	196,8	224,8	190,5	192,5	192,7	170,0
„ „ in metres per sec.	0,956	1,092	0,912	0,937	0,757	0,810	1,000	1,142	0,968	0,978	0,979	0,864
IP	89,14	116,00	71,75	74,60	47,24	54,80	85,80	110,50	77,06	77,50	55,93	46,40
Steam used per IP per hour by direct measurement in ℔s engl.	26,247	23,15	29,616	24,09	35,075	27,11	23,21	20,36	23,036	20,71	23,765	25,11
„ „ „ „ „ „ „ „ „ in kilos	11,90	10,50	13,43	10,92	15,91	12,29	10,53	9,23	10,45	9,39	10,78	11,39
„ „ „ „ „ „ „ the cards **) in ℔s engl.	17,352	16,25	17,755	15,58	21,028	16,42	12,346	15,25	12,312	15,76	12,699	18,53
„ „ „ „ „ „ „ „ „ in kilos	7,87	7,37	8,05	7,06	9,54	7,45	5,60	6,92	5,58	7,15	5,76	8,40
„ lost „ „ „ „ in ℔s engl.	8,895	6,90	11,861	8,51	14,047	10,69	10,864	5,11	10,724	4,95	11,066	6,58
„ „ „ „ „ „ in kilos	4,03	3,13	5,38	3,86	6,37	4,85	4,93	2,32	4,96	2,24	5,02	2,98
Percentage of steam lost to total steam used	33,89	29,80	40,05	31,17	40,05	39,43	46,81	25,09	46,55	23,90	46,56	26,20
Consumption of coal per IP per hour in ℔s engl.	2,87	2,53	3,24	2,64	3,84	2,97	2,54	2,23	2,52	2,27	2,60	2,75
„ „ „ „ „ „ „ in kilos	1,29	1,15	1,47	1,19	1,74	1,35	1,15	1,01	1,14	1,03	1,18	1,24

*) Engineering 1875. I. P. 14.

**) The figures in this line are referred to the L. P. card.

Table*) of the steam losses in the jacketed compound engine of the steamer "Rush".

1	2	3
Duration of the experiments in hours	55,00	6,00
W. P. of boiler in lbs per □" engl.	69,06	36,73
" " " " kilos per □ cm	4,85	2,58
Ratio of cut-off	0,143	0,182
" " expansion including clearance	6,216	4,030
Piston speed in feet per min. engl.	318,60	249,70
" " " metres per sec.	1,53	1,27
IHP	266,55	168,65
Steam used per IHP per hour by direct measurement in lbs engl.	18,38	22,09
" " " " " " " " " in kilos	8,30	9,98
" " " " " " " the cards**) in lbs engl.	13,52	16,94
" " " " " " " " " **) in kilos	6,13	7,68
" lost " " " " in lbs engl.	4,86	5,15
" " " " " " in kilos	2,20	2,34
Percentage of steam lost to total steam consumed	26,41	23,30
Consumption of coal per IHP per hour in lbs engl.	2,43	—
" " " " " " " in kilos	1,102	—

of the jacket as saturated steam can, for superheated steam is only then condensed on them when it has first cooled down in their immediate neighbourhood to its saturation temperature. Besides, a given volume of superheated steam passing through in a unit of time contains by § 13, 11 less heat than an equal volume of saturated steam.

Boiler-steam.

18) d. **The heating of the cylinder with boiler-steam,** taken to the jackets by a branch pipe connection from the main steam pipe, is the arrangement almost universally adopted in the marine engines of to-day. Its effect upon the process in the cylinder has been already described in § 15, 22 to 25. The experiments carried ont by EMERY under the direction of Engineer LORING of the United States Navy in 1874 and 1875 on five different American Revenue Cruizers are the best proof of the reduction of steam losses in surface condensing engines attained by the use of jacketing with boiler steam.

American experiments.

19) The arrangement and dimensions of cylinders of these engines are given in the table on P. 84. All the experiments reported in the tables took place at moorings, those of the "Bache" at Baltimore, the others at Boston. The "Bache's" engine was a

*) Engineering. 1875. I. P. 129.

**) The figures in this line are referred to the L. P. card.

tandem and was arranged so as to work with the *IP* cylinder disconnected, or with both cylinders receiving steam independently as a single expansion engine. The cylinders of the "Rush's" engines were placed as usual, beside each other.

20) It follows from the foregoing tables that the loss of steam, amounting to between 30 and 40 % of the total steam consumed in the single cylinder engine and also in the Woolf's engine, worked without the jacket, is reduced to from 20 to 30 % on an average when the jacket is used. *So that in general an economy of steam of at least 10 % may be obtained by jacketing with boiler steam in condensing engines.* This economy would be still greater if the steam could traverse the jacket in a current, instead of remaining at rest in it as in these American engines and in most marine engines of the present time. But the most modern triple-expansion engines, if they are jacketed at all, are often so arranged that the steam traverses the jacket for the reason mentioned in 13). The heat of the steam passing off from the jacket is used for evaporating sea water, the vapour of which, being led into the condenser produces distilled water for the supplementary feed, instead of salt water being used. The condensed water which collects in the jacket flows into the hot-well and is used as feed, while the last residue of the steam serves to heat the feedwater. **Inferences from the American experiments.**

21) e. **Heating the cylinder with steam hotter than that used in the engine** has been shewn by all experiments made with it to be particularly advantageous. Of all these experiments let us select three carried out at different dates and in different countries. They are those of EMERY, DELAFOND, and GUZZI. **Jacket steam hotter than the steam used in the engine.**

22) *Emery's experiments**) took place in 1875 on board the steamer "Gallatin" at Boston. The boiler steam was kept at a working pressure of nearly 5 kilos per □cm and admitted to the jacket, while it was throttled down to about 1 kilo per □cm on reaching the cylinder. The condensed water was blown out of the jacket from time to time. **Emery's experiments.**

23) *Delafond's experiments***) were carried out in 1884 at SCHNEIDER's works at Creuzot on a stationary Corliss engine with one cylinder 55 cm diar. and 110 cm stroke which was only jacketed at the sides. The steam passed from the boiler into the jacket at 7 k per □cm working pressure, but was reduced in the **Delafond's experiments.**

*) Engineering 1876. I. P. 124.

**) Annales industrielles — Feb. 1st. to Mar. 22nd. 1885.

Table of the economy obtained by using hotter than boiler steam in the jacket.

Experiments carried out by	Emery in America 1875			Delafond in France 1884			Guzzi in Italy 1886	
1	2	3	4	5	6	7	8	9
Conditions of jacketing	jacket not used	boiler steam in jacket	hotter steam in jacket	jacket not used	boiler steam in jacket	hotter steam in jacket	boiler steam in jacket	hotter steam in jacket
Duration of the experiments in hours . .	2,25	2,13	2,20	—	–	—	7,18	6,30
Cylinder pressure in kilos per □ cm . .	0,90	0,92	0,98	3,79	—	4,15	3,95	3,98
Jacket " " " " " . .	—	0,92	4,92	—	boiler-steam	7,00	3,95	12,40
Admission temperature in the cylinder in °C	117,96	118,28	119,24	149,40	—	152,10	150,56	150,76
Temperature of the jacket steam in °C .	—	118,28	157,42	—	—	169,46	150,56	192,00
Difference of temperature of steam between cylinder and jacket	—	0	38,18	—	0	17,36	0	41,24
Ratio of cut-off	0,64	0,65	0,63	0,57	—	0,57	—	—
IP	90,17	97,87	102,91	133,70	—	162,80	25,67	25,90
Steam used per *IP* per hour in kilos . .	20,01	16,96	15,81	10,55	—	10,00	10,66	8,89
Economy, expressed as a percentage of total steam used when jacket not in operation	—	15,24	20,98	—	2,8	5,2	—	16,60*)

*) This number expresses the economy as a percentage of the total steam used when the jacket is heated with boiler steam.

cylinder to only about 4 k per □ cm by throttling. The jacket was kept clear of condensed water by an automatic arrangement.

24) *Guzzi's experiments**) were made in February 1886 on a stationary engine in Italy. He took his jacket steam at 12.4 k per □ cm from a very small tubulous boiler of similar construction to Perkins's (see further on, under "water-tube boilers"). This small boiler was placed in the furnace of the large boiler which supplied the engine at a working pressure of 4 k per □ cm. The jacket drain was also blown into the large boiler. Guzzi's experiments.

25) The above table is compiled from the results of these experiments, clearly shewing the economical action of the jacket in all cases where it is filled with steam which is hotter than the main steam. Here also the results obtained would have been still better if the steam had not, in every case, remained at rest in the jacket instead of continuously passing through it in a current. Diminution of the advantages.

26) The table shews in general *that by using steam in the jacket about 40° hotter than the cylinder steam, an economy of steam is obtained about 20 % above that of an unjacketed cylinder and 10 %* (see 20) *above that of a cylinder jacketed with main steam.* But as at high pressures, such a large difference of temperature between the jacket-steam and the cylinder-steam cannot well be attained without considerable increase of risk, it has been proposed to heat the jacket with the vapour of linseed oil instead of steam, as this oil does not boil under atmospheric pressure till heated to 370°. Inferences from the experiments.

27) **III. Losses of steam in the jacket.** The transition of heat from the jacket-steam to the cylinder wall causes condensation and therefore a certain loss of steam, the magnitude of which has been variously determined. As shewn by the experiments of LONGRIDGE, DELAFOND and DOERFEL, quoted below, the loss by condensation in the jacket is always slight compared with that caused by initial condensation in the cylinder, otherwise the jacket would be of no use. Losses of steam in the jacket.

28) *Longridge's experiments***) were made in the autumn of 1881 with a compound condensing stationary engine at Blackburn. The cylinders were 50.8 cm and 86.3 cm diar. and 152.4 cm stroke. The cylinder sides and the receiver were jacketed, but not the cylinder ends. The three jackets could be shut Longridge's experiments.

*) Engineering 1888. I. P. 222.

**) Engineering 1882. I. PP. 174, 220, 242, and 266.

Table of the steam used in the jackets of a compound engine.

Average duration of each experiment	Without jacketing.	Without jacketing.	Only HP cylinder jacketed.	Only the receiver jacketed.	Only LP cylinder jacketed.	All jackets used.	All jackets used.
1	2	3	4	5	6	7	8
Working pressure of boiler in ℔s per □" engl.	89,4	88,8	89,0	88,9	88,4	89,8	87,8
„ „ „ „ in k per □ cm	6,28	6,24	6,25	6,24	6,21	6,31	6,17
Ratio of cut-off in *HP* cylinders including clearance	0,294	0,293	0,279	0,275	0,272	0,235	0,285
Total ratio of expansion	8,27	8,29	8,58	8,71	8,78	9,82	8,49
Piston-speed in feet per min.	472,4	478,9	470,2	474,8	473,8	474,6	487,0
„ „ „ metres per sec.	2,400	2,433	2,394	2,412	2,407	2,411	2,474
IHP	313,62	314,29	313,48	318,67	314,63	313,78	338,02
Total steam used per *IHP* per hour in ℔s engl.	16,87	16,97	16,26	17,29	17,54	17,00	17,16
„ „ „ „ „ „ „ „ kilos	7,65	7,69	7,37	7,84	7,95	7,71	7,78
Steam used in *HP* jacket during the whole experiment in ℔s engl.	13,30	5,00	827,50	40,00	10,00	752,50	780,00
„ „ „ „ „ „ „ „ in kilos	6,03	2,27	375,35	18,14	4,53	341,33	353,80
„ „ receiver jacket „ „ „ „ in ℔s engl.	10,00	17,50	170,00	970,00	10,00	990,00	900,00
„ „ „ „ „ „ „ „ in kilos	4,53	7,94	7,71	440,00	4,53	449,06	408,24
„ „ *LP* jacket „ „ „ „ in ℔s engl.	10,00	10,00	120,00	10,00	1057,50	1245,00	1140,50
„ „ „ „ „ „ „ „ in kilos	4,53	4,53	54,43	4,53	479,68	564,73	517,33
„ „ all three jackets „ „ „ „ in ℔s engl.	33,30	32,50	1117,50	1020,00	1077,50	2987,50	2820,50
„ „ „ „ „ „ „ „ „ in kilos	15,10	14,74	506,89	462,67	488,75	1355,13	1279,37
„ „ „ „ „ per *IHP* per hour in ℔s engl.	0,014	0,013	0,467	0,419	0,450	1,246	1,939
„ „ „ „ „ „ „ „ „ in kilos	0,0063	0,0059	0,2113	0,1900	0,2041	0,5651	0,8795
Percentage of the jacket steam to total steam used, as measured	0,08	0,07	2,87	2,42	2,56	7,33	11,30
Coal consumed per *IHP* per hour in ℔s engl.	1,88	1,96	1,86	1,89	1,92	1,85	2,04
„ „ „ „ „ „ in kilos	0,85	0,89	0,84	0,85	0,87	0,84	0,92

off from each other by stop-valves, so that each could be used separately. But these stop-valves were not quite tight, which explains the consumption of steam in the jackets that were disconnected. The table on P. 92 gives an abstract of the results of these experiments and shews that the steam used in the jackets was 7.33 to 11.3 % of the total.

29) *Delafond's experiments*, which took place at Creuzot in 1884 are described in abstract in the following table, shewing that the saving of steam due to the jacket is always a multiple, generally even a very considerable multiple, of the steam consumed in it. **Delafond's experiments.**

Table of the steam used in the jacket of a single expansion engine.

Conditions of working	Engine worked: Noncondensing			Condensing		
1	2	3	4	5	6	7
Working pressure of boiler in k per □ cm	7,75	5,50	3,50	7,75	5,50	3,50
Steam used per IP per hour in kilos, without jackets	14,94	13,73	13,60	11,97	11,00	10,64
Steam used per IP per hour in kilos, with jackets	12,07	12,37	12,64	9,72	10,12	10,43
Saving due to use of jacket per IP per hour in kilos	2,87	1,36	0,96	2,25	0,88	0,21
Steam used in jacket per IP per hour in kilos	0,302	0,210	0,139	0,292	0,304	0,147
Weight of steam saved was the following multiple of the weight of steam used in jacket	9,5	6,5	6,9	7,7	2,9	1,4

30) *Doerfel's experiments* were carried out in 1885 with a two cylinder compound Corliss engine 56 cm and 84 cm diar. × 120 cm stroke, made by the Prague Engineering Co. for a spinning mill. According to OTTO W. MÜLLER jun.*) the steam used in the jacket was 7 % of the total at 240 *IP*. **Doerfel's experiments.**

31) From the foregoing figures we may conclude that *the steam used in the jackets of well designed compound engines will average about 7 to 10 % of the total.* If it rises, as may well occur under unfavourable circumstances, to from 15 to 20 % the advantage of the jacket is very considerably diminished. The consumption of jacket steam in "triples" is given in 47). **Inferences from these experiments.**

32) **IV. Reduction of the advantage of jacketing.** The economical effect of jacketing may be surprisingly reduced, in fact practically vanish, if **Reduced advantage of jacketing.**

*) Zeitschrift des Vereines deutscher Ingenieure 1887. P. 526.

a) the cylinder is very large,
b) the piston-speed is very high,
c) the cut-off is very late,
d) the range of temperature is very small,
e) the cylinder is very copiously lubricated,
f) air is mixed with the admission steam,
g) the jacket steam has a low temperature,
h) the freqent blowing out of the condensed water in the jacket is neglected.

Size of the cylinder.

33) a. **Influence of the size of the cylinder.** The greater the diameter of the cylinder, the smaller is the heating surface of the jacket in proportion to the enclosed volume of steam (in the cylinder). A cylinder of 0.5 m diar. and 1 m stroke, for instance, has a jacket surface of $0.5 \times \pi \times 1 = 1.57 \square$ m + the cover and bottom surface of $\frac{\pi}{4} 0.5^2 = 0.196 \square$ m, which gives a total internal surface of 1.766 $\square$ m. The volume of steam occupying the cylinder, and always being cooled at one end by the piston, is 0.196 cbm, so that per cubic metre of steam there are at the rate of 9.01 $\square$ m internal surface. But if the cylinder is 3 m diar. and 1 m stroke (as before), its total internal surface is 16.5 $\square$ m, its contents 7.07 cbm, and it has only 2.33 $\square$ m of internal surface per cbm of volume of steam. *The jacket of the small cylinder is therefore* $9.01 : 2.33 = 3.85$ *times as efficient as that of the large one.*

Piston-speed.

34) b. **Influence of the piston-speed.** If the piston-speed of the small cylinder is only 1 m per second, as was usual in the older engines, the cylinder is filled exactly once in a second, and as just shewn, there are 9.01 $\square$ m of internal surface per cbm of steam per second. But if the large cylinder has 3 m piston-speed, as quite customary in modern engines, then the enclosed steam is changed three times in a second, so that there are only 0.77 $\square$ m of internal surface per cubic metre of steam per second. *The jacket of the small cylinder with its low piston-speed is therefore* $9.01 : 0.77 = 11.8$, *or in round numbers 12 times, as efficient as that of the large cylinder with the higher piston-speed.* The same conclusion may be drawn from the table on page 86 referring to the "Gallatin" experiments. While the losses here at little more than 1 m piston-speed were 36 % without the jacket, against 22 % with it, i. e. 14 % in favour of the jacket, at $1\frac{1}{2}$ to $1\frac{3}{4}$ m piston-speed this difference came out only $29 - 20 = 9$ % in favour of the jacket, although the initial pressure was about four times as high and the cut-off nearly three times as early in the latter

case as in the former, and therefore also the difference of temperature (between the cylinder wall and the admission steam) to be equalized by the jacket steam was much greater.

35) **c. Influence of the ratio of cut-off.** Most modern single expansion marine engines have an expansion slide which governs the period of admission while the main slide regulates that of the exhaust. As the main slide is, or ought to be, always at full gear, it gives the same duration of exhaust for all possible positions of the expansion slide, or what is the same thing, an equal period of injurious re-evaporation. If a certain weight of steam is condensed (in a cylinder) it will go on gradually increasing till it reaches the amount which can just be re-evaporated during the exhaust. As, by the foregoing, the amount of re-evaporation may in general be regarded as constant, it follows that the loss due to condensation in the cylinder becomes the more sensible, the earlier the cut-off is made, as shewn in § 16, 7 and proved by D. K. CLARK*) in 1852 in his experiments on locomotives. *So that for larger ratios of cut-off the loss will become a smaller percentage of the total steam used and the action of the jacket of the less advantage.* Ratio of cut-off.

36) **d. Influence of the range of temperature.** In single expansion engines with a working pressure of 6 k per □ cm the steam entered the cylinder at 160° and escaped to the condenser at 60° to 80°. The difference between the admission and release temperatures, or the range of temperature, was therefore about 80° to 100°. In the *HP* cylinder of the newest triples working at 10 to 12 k per □ cm (150 to 180 ℔s) the steam enters the *HP* cylinder at about 180° to 190° and leaves the *LP* cylinder at 60° to 80° as before. Thus the range of temperature, evenly distributed over the three cylinders, only comes to about 30° to 40° for each cylinder and consequently the cylinder wall is not cooled down during a revolution to so great an extent as in the former case. Without jacketing, the loss by condensation in the cylinder must be much greater in the single-expansion engine than in the triple with the much smaller range of temperature. From these considerations it is clear *that the jacket in the modern fast-running multiple-expansion engines no longer possesses the importance it had in slow running single-expansion engines, and that it may perhaps again be completely dispensed with if, as seems possible, the number of expansions as well as the piston-speed is still further increased in the future.* Range of temperature.

*) D. K. CLARK. Railway machinery. London 1852.

Longridge's experiments.

37) *Longridge's experiments* with the compound engine confirmed the above conclusion, for they shewed (see Table P. 92) that it was not only almost indifferent, so far as consumption of steam was concerned, whether the engine worked with the jacket in use or not, but that there was even less steam used when the jackets were shut off altogether. If we cannot exactly infer from this that jackets in compound engines are in all cases valueless, we may at any rate recognize that the steam losses by condensation in the cylinders, by the passage of the steam from the *HP* to the *LP* cylinder, and by the jacket consumption may, when all three are taken together, amount to as much as the loss in unjacketed cylinders.

Weight of steam condensed.

38) In these experiments the weight of steam condensed per □ m of the wall surface was ascertained to be as follows:

Table of the weight of steam per □ m of wall-surface in Longridge's compound engine.

Description of the surface	Area of surface in □ m	Internal temperature °C	External temperature °C	Difference of temperature °C	Weight of steam condensed kilos	Quantity of heat passed though in thermal units.
1	2	3	4	5	6	7
Steampipe	11,33	164,5	32,8	131,7	0,0371	3,72
HP steamchest	1,07	164,5	21,7	142,8	0,0039	0,44
HP cylinder cover and bottom (not jacketed)	0,41	132,2	21,7	110,5	0,0009	0,11
HP cylinder sides (jacketed) . . .	2,51	164,5	21,7	142,8	0,0088	0,88
Communication pipe and receiver cover and bottom (not jacketed)	1,56	110,5	30,0	80,0	0,0029	0,33
Receiver sides (jacketed)	2,77	164,5	21,7	142,8	0,0093	0,94
LP steamchest (not jacketed) . .	1,24	110,0	21,7	88,3	0,0015	0,16
,, cylinder cover and bottom (not jacketed)	1,17	78,9	21,7	57,2	0,0015	0,16
LP cylinder sides (jacketed) . . .	4,22	164,5	21,7	142,8	0,0156	1,77

Rankine's assertion.

39) From the table on P. 92, shewing the consumption of steam per *IHP* with only the *HP* cylinder jacketed to have been 7.37 kilos and with only the *LP* cylinder jacketed 7.95 kilos, LONGRIDGE draws the conclusion that if it were desired to jacket only *one* cylinder of a compound engine in order to save expense or weight, this should be the *HP* cylinder. In this he is in accord with RANKINE*) who states that, in certain compound engines of which only the *HP* cylinders were jacketed, no appreciable internal condensation was found.

*) W. J. M. RANKINE. A manual of the steam engine. London 1873. P. 396.

Thence he (RANKINE) infers that the steam, passing from the *HP* to the *LP* cylinder, receives a certain quantity of heat from the *HP* cylinder, either directly, or indirectly by conduction through the *LP* cylinder, and is thus protected from cooling and condensation. But considering that we are now richer by the experience of sixteen years than RANKINE was, and bearing in mind the figures of LONGRIDGE'S last table, as well as the results of EMERY and DONKIN & SALTER'S experiments, the above conclusion is no longer tenable.

40) LONGRIDGE'S *single* experiment, just quoted, cannot be regarded as convincing, because the jacket stop-valves were not tight. During the experiment with the *HP* cylinder jacketed, about 5 kilos of steam passed into the *LP* jacket which was supposed to be shut off; and vice versâ, about 50 kilos, or ten times as much, leaked into the *HP* jacket during the experiment which was intended to be made with only the *LP* cylinder jacketed. Further, the range of temperature was less in the *LP* cylinder than in the *HP*, whereas in most compound engines the contrary is the case. And finally — the covers and bottoms of both the cylinders and receiver were unjacketed. Besides all this there is the important circumstance that such steam as is condensed in the *HP* cylinder is for the most part re-evaporated in it during the exhaust, and passing to the *LP* cylinder, performs useful work there, at a lower pressure certainly, but acting on a much larger piston surface; whereas the steam arising in the same manner by re-evaporation in an unjacketed *LP* cylinder passes into the condenser and is lost. **Refutation of Longridge.**

41) *From Emery's experiments* (see table PP. 86 to 89), we see that according to the results of the Woolf engine of the "Bache" and the compound engine of the "Rush", the unjacketed *HP* cylinder of the former diminished its economy as much as the jacketed *HP* cylinder of the latter increased its economy. A comparison of the completely jacketed compound engine of the "Rush" (see table, P. 89 column 2) with the unjacketed single-expansion engine of the "Dexter" (see table, P. 86 column 8) shews, under similar circumstances, a saving of steam in favour of the first of 22.94 $^0/_0$; whereas the "Bache's" engine working as a "Woolf" (see table, P. 88 col. 9) with only the *LP* cylinder jacketed attained to an economy of 22.54 $^0/_0$ more than as a single-expansion engine, i. e. working with the *LP* cylinder by itself and its jacket not in use (see the same table, col. 2). Further, the "Bache's" engine worked as a Woolf engine with the *LP* cylinder jacketed (see table 88, col. 9), shews a saving of **Emery's experiments.**

steam of 12.19 % over its results as a single-expansion engine with the *LP* cylinder likewise jacketed (the same table, col. 3); whereas the compound engine of the "Rush" with both cylinders jacketed gives an economy of 12.65 % over the "Dexter" (supposing the latter to have attained a similar saving when jacketed as the "Bache" did). So that the advantage which the "Rush's" engine with jacketed *HP* cylinder possesses over the "Bache's" with the same cylinder unjacketed, amounts to 12.65 — 12.19 = 0.46 %. We may therefore infer from both comparisons that the jacketed *HP* cylinder of the "Rush" was economically scarcely at all superior to the unjacketed one of the "Bache", hence it is quite justifiable to make the *HP* cylinder without a jacket.

Donkin & Salter's experiments.

42) *Donkin and Salter's*) experiments* were made with a Woolf's engine of 15.2 cm and 25.4 cm diar. × 30.4 cm stroke erected at their works at Bermondsey, and fitted with all necessary appliances for experimental purposes. According to LÜDERS**) this firm have been in the habit for the last twenty years of carrying out scientific experiments with the engines they have built. DONKIN and SALTER employ thermal measurement in their trials, determining by exact observation of the weights and temperatures of the injection water and condensed water, the number of thermal units rejected per *IHP* per hour. A comparison with the quantity of heat supplied to the engine per *IHP* per hour then gives a measure of the efficiency (see § 17). As the losses by radiation &c. are always the same for the same engine, this method of experimenting is pretty accurate.

Inferences from the experiments.

43) The following table contains the particulars of the best two experiments of each series, i. e. the two in which the least heat was lost. The mean values in the last line but one are the arithmetical means of the whole series which comprises, e. g. for working with jacketed *LP* cylinder, 82 separate experiments. Each trial lasted half an hour, during which the heat quantity rejected in the condensed water was measured every minute. As the results shew, 419 thermal units were rejected per minute on an average when only the *HP* cylinder was jacketed, whereas this quantity was but 400 with only the *LP* cylinder jacketed, proving the latter mode of working to be nearly 5 % more economical. So that if it is a question

*) Engineering 1886. II. PP. 487 & 577.

**) Zeitschrift des Vereines deutscher Ingenieure 1882. P. 239.

of jacketing only one cylinder of a compound engine, *it is decidedly more advantageous to jacket the LP than the HP cylinder.*

Table of Donkin & Salter's experiments with a Woolf engine.

Conditions under which the experiments were conducted	jackets not in use		only the *HP* jacket in use		only the *LP* jacket in use		both jackets in use	
	—		the steam entered the jacket and passed from it through the stop-valve into the *HP* steam-chest.		the steam traversed the jacket.		the steam passed first through the *LP* jacket, then through the *HP* jacket then through the stop-valve into the *HP* steam-chest.	
1	2	3	4	5	6	7	8	9
Working pressure in k per □ cm	3,00	3,00	3,00	3,00	2,80	2,80	2,90	3,10
IHP	4,91	5,79	8,84	8,03	8,72	9,82	6,50	7,01
Revolutions per minute	98,17	96,61	97,50	97,93	106,80	101,30	93,32	103,40
Ratio of expansion	8,38	7,54	7,01	7,73	9,74	8,38	13,10	13,10
Heat quantity*) rejected in the condensing water in thermal units	525	533	418	420	380	383	347	348
Mean of these heat quantities for all the experiments of the respective series	563		419		400		378	
Water collected from jackets per *IHP* per hour	—	—	0,024	0,029	0,036	0,029	0,049	0,049

44) *Jacketing the HP cylinder.* Modern practice is invariably in accord with the experience thus gained, for the *HP* cylinders of compound, as well as triple and quadruple expansion engines working at high pressures with late cut-off and considerable piston-speed, have been lately, for the most part, simply well cleaded but not jacketed, because it is desirable to retain a certain degree of moisture of the steam in them. The moisture of the steam lubricates the smooth surfaces of the slides and cylinders and keeps the packing wet, thus allowing less cylinder oil to be used. This is of very little service in any case, because on the one hand it is too rapidly volatilized in consequence of the high temperature of the steam, and on the other hand, on passing into the boilers with the feed water, it is deposited under certain circumstances as an insoluble

Jackets for *HP* cylinders.

*) These thermal units are English; to convert them into those of the metrical system the above figures are to be multiplied by 0.555.

soap on the crowns of the furnaces, conducing, as will be explained later, to the risk of their coming down. — This lubrication by the steam itself, which is probably due to the attraction between the watery particles and the metal, was always quite enough for the jacketed *HP* cylinders of the older compound engines with their lower working pressures, but it does not appear to be present in a sufficient degree in the *HP* cylinders of modern high-pressure triple-expansion engines, if these are jacketed, so that it is preferable not to jacket them. — It may be remarked that RANKINE*) also points out the desirability of lubricating the cylinder by means of the moisture of the steam.

Jacketing of the *MP* cylinder.

45) *Jacketing the MP cylinder.* Some engineers do not fit a jacket to the *MP* cylinder, while others consider it indispensable. It is probably advisable to jacket the *MP* cylinder where the weight of the engine is not closely limited. But an unjacketed *MP* cylinder can never have any very sensible effect on the steam losses as (by 40) the re-evaporation taking place in it is turned to useful account in the *LP* cylinder.

Jacketing of the *LP* cylinder.

46) *Jacketing the LP cylinder.* The *LP* cylinder is usually jacketed; for its range of temperature is in nearly all engines the greatest, so that the difference between the temperature of the steam at admission and that of the cylinder walls (and therefore also the internal condensation) is of the most importance. The considerable quantities of re-evaporated steam, which in the unjacketed cylinder are not produced till during the exhaust, thus pass to the condenser without doing work and are lost. The jacketing of the *LP* cylinder is therefore under all circumstances to be recommended. The jacket is supplied with steam either from the receiver of the next cylinder, or a branch from the main steam pipe is connected with a reducing-valve to the jacket, to which the steam is admitted at a pressure corresponding to the admission pressure in the *LP* cylinder. But according to EMERY, DELAFOND, and GUZZI's experiments (see 25) it would greatly conduce to increased efficiency of the *LP* jacket to pass main boiler steam through it, provided it were designed of sufficient strength.

Jacketing dispensed with.

47) In "triples" with *very high* piston-speed and *very late* cut-off, where the smallest possible weight of machinery is an object, as in torpedo-boats, the *LP* jacket is dispensed with, experience shewing that the steam losses thus induced are not indeed particularly noticeable. A considerable number of triple

*) W. J. M. RANKINE. A manual of the steam engine. London 1873. P. 396.

and quadruple expansion engines for cargo-steamers, having piston speeds of 2 to 2.5 m per second and a cut-off in each cylinder of about 0.6, have recently been constructed in England without jackets at all and have given very good economical results compared with the ordinary compound engine with jackets. According to this, KIRK appears to have been, even at that time, quite right when he asserted in 1882 after the trials of the S. S. "Aberdeen" with the first successful triple expansion engine, that her steam losses with the jackets not in use would not exceed the losses in the cylinders with the jackets in use plus the steam condensed in the jackets. In the "Aberdeen" the *HP* cylinder was unjacketed while the other two were jacketed. The initial effective pressure of the *MP* engine was 2 k per □ cm and that of the low 0.66 k. From careful measurements on the trial-trips KIRK determined the weight of steam used in both jackets at 3.75 % of the greatest weight shewn on the *HP* cards. We shall therefore be not far wrong if *we estimate the weight of steam used under favourable conditions in the jackets of well designed triples at 5 % of the total.*

48) e. **Influence of lubricating the cylinder.** Simultaneously with their experiments referred to in 42) DONKIN & SALTER investigated the value of cylinder lubrication and found that the loss of heat in their Woolf's engine with both cylinders very copiously lubricated and the jackets not in use, was nearly 2 % less than when the jackets were used but the cylinders received no oil at all. This is explained by the abundant lubrication covering the cylinder walls with a layer of grease which at first hinders the rapid transmission of heat to them from the entering steam, thus reducing the quantity of water precipitated on them, and afterwards favours the radiation from the walls so that the re-evaporation is almost completed during the expansion period. On this account it is probable that the almost universal, utterly extravagant, oiling of the cylinders on trial trips is to a certain extent to blame for the same economical results never being afterwards obtained in ordinary working. An abstract of the particulars of these very interesting experiments is here omitted, because it is desirable to limit the lubrication of the cylinders as much as possible, for reasons explained in 44), so that the economy of steam obtained by this means, quite apart from its expense, is of no practical importance.

Lubrication of the cylinder.

49) f. **Influence of aërating the steam.** As already mentioned in § 14, 7 (P. 66), condensation on cool cylinder-walls is considerably

Aërated steam.

diminished by the presence of air, so that this must also reduce the efficiency of the jacket. The question of the practical advisability of pumping air into high-pressure steam before its admission into the cylinders of marine engines was discussed in § 14, 11. As it was there shewn that aërated steam can only be applied to non-condensing engines, the experiments briefly referred to in § 14 will not be further treated of here.

Temperature of the jacket steam.

50) g. **Influence of the temperature of the jacket steam.** The cooler the jacket steam, the lower is the mean temperature of the walls and therefore the greater the condensation during admission and the re-evaporation during exhaust. The jacket is most useful when supplied with *hotter* steam than the engine itself (see table, P. 91) and will therefore always shew less favourable results with *cooler* steam.

Draining of the jacket neglected.

51) h. **Influence of neglecting to drain the jacket.** The condensed water collecting in the jacket must be blown out from time to time into the condenser or hot-well. If it remains standing in the jacket the cylinder is not only no longer heated, but a large amount of heat is continually being abstracted from it by this water. A cylinder whose jacket is filled with water therefore works with greater losses than a cylinder without a jacket at all. Besides which, any water left in the jackets after the engines are stopped prevents the heating of such portions of the cylinders as it covers, when they are next "warmed through", thus producing unequal expansion of the cylinders and jackets which has been a fruitful cause of cracked cylinders.

Water-gauges for jackets.

52) Unfortunately the accumulation of water in the jackets through neglect of the engineers to blow them off is in KIRK'S*) experience a very common occurrence. In order to reduce the chance of it, a water-collector furnished with a gauge glass has been fitted to inverted engines at some handy and easily visible spot as low as possible on the jacket, so that the formation of water can be constantly observed. Such an arrangement is however of very little use in horizontal engines as it is low down in the bilge and therefore invisible, which is the reason of the jacket drain being most neglected in these engines. How much importance is attached to this circumstance is evident from the fact that many engineers never jacket horizontal compound or triple-expansion engines because they will not endanger the reputation of their firms by the possibility of neglect in the engine-room, and therefore, aban-

*) Transactions of the Institution of Naval Architects 1882. P. 50.

doning from the first the idea of exceptional results, are satisfied with a lower average performance.

53) For inverted engines automatic jacket-drains have quite recently come into use, some of which are stated to work well, for instance those depending on the expansion by heat of a brass tube as used by THORNYCROFT in his torpedo-boats, and described later in the division on cylinders. **Automatic jacket drain.**

54) The conclusion to be drawn from all the foregoing investigations and further borne out by certain experiments in Alsace, BORODINS'*) locomotive trials, &c., is as follows, viz that the value of the jacket is the smaller, the greater the diameter of the cylinder, the piston-speed, and the cut-off, the smaller the range of temperature, the greater the quantities of lubricant and air admitted to the cylinder, the cooler the jacket steam, and the greater the intervals of time allowed to elapse without clearing the jacket of condensed water. The value of the jacket is therefore in many cases so small that the engineers are quite right to shut it off at sea and only use it for warming through. In such cases the only advantage of the jacket is that it affords the designer an opportunity of putting a liner in the cylinder of a harder mixture than the rest of the casting. **Conclusion from all the investigations**

55) The following Rule for multiple-expansion engines of the present day is based on the results of universal experience. *If the greatest economy of steam is desired, the low-pressure cylinder should always if possible be fitted with a jacket supplied with direct main steam, whenever the limit of weight of engine allows the jacket to be made sufficiently strong. The steam must traverse the jacket; after which its remaining heat should be used for warming the feed or in a supplementary-feed evaporator.* **Rule for multiple-expansion engines.**

§ 17.

Heat-trials of marine engines.

1) **I. Object and method of conducting heat-trials.** The object of the calorimetric investigation of the action of the steam engine (first applied by HIRN) is to ascertain by practical experiment with the engine in actual work, the thermal influences of the cylinder walls described in § 15 which cannot be theoretically determined with precision, for want of trustworthy coefficients for the heat-conductivity of iron and the transmission of heat from iron to steam and vice versâ, as well as on account of other varying circumstances affecting **Object.**

*) Engineering 1886. II. PP. 248, 273, 301 and 328.

the results, and if possible to reduce these thermal influences to numerical values of universal applicability to different types of engines under similar conditions of working.

Carrying out the trials.

2) Trials for determining the efficiency of engines, by measuring the heat supplied in the steam and the heat used, can be made comparatively easily and with tolerable accuracy on stationary jet-condensing engines. With marine engines the difficulties are greater and the attainable degree of precision lower. In all these experiments the principal object is to determine the "heat supplied" to the engine per stroke in the steam and the "heat rejected" in the condensing water, as well as to observe the internal changes taking place in the cylinders &c.

Difficulties of marine engine trials.

3) The determination of the heat supplied, as we shall see below, is not very difficult. The simplest way of measuring the heat rejected in stationary engines, is to lead the airpump discharge into a rectangular trough in which a weir is fitted, and to measure from time to time the depth, temperature, and velocity of the water escaping over it. This arrangement cannot be used at sea, apart from the motion of the vessel, even if the engine has a jet-condenser, and is obviously of no use with a surface condenser, as it would leave the heat carried off by the circulating water out of account. Supposing this heat quantity to be approximately ascertained by observing the temperature and velocity of efflux of the circulating water, the figures cannot be checked as there are no means of measuring the circulating water with accuracy. If the circulating pump is a piston pump of ascertained efficiency the water may be measured with some approach to exactness, but it is unfortunately very difficult to keep a pump working steadily under the same conditions for any length of time. With a centrifugal circulating pump the measurement of the water must be still more inaccurate and therefore the value for the heat rejected still less trustworthy.

Checking the observed quantities.

4) For the above reasons it is impossible to check the heat rejected in a marine engine directly. But if a long series of trials are conducted with the most scrupulous care and those referring to the same *IHP* under different circumstances are grouped together, we are enabled, by compiling the results graphically, to recognize and reject incorrect observations. The others can then be further checked by comparing them with the accepted results of stationary engines of the same type, so that in this way the heat-trials of marine engines may lead to practically useful conclusions.

5) **II. Evolution of the principal equations.** According to GRASHOF *) the heat-movement in the cylinder of a single-expansion condensing engine may be divided into the four periods of admission, expansion, exhaust, and compression, assuming that the piston and slide are perfectly tight and that there is no lead either on the steam or exhaust side. The mixture of steam and water in the cylinder at the end of each of these four periods is to be represented by the magnitudes distinguished by the marks 1, 2, 3 and 0 on Fig. 5, Plate 2. Periods of the stroke.

6) **Process during the admission period.** One kilo of the mixture of steam and water supplied by the boiler at temperature t and pressure p, which latter, neglecting resistances in the steam pipe, may be taken as equal to the boiler pressure, may lose by cooling on its way to the cylinder an extra heat quantity Q_x. If it then has the dryness-fraction x_x, the total heat supplied to the 1 kilo from water at 0^0 is Heat of the admission steam.

$$\lambda = q + x_x \varrho \text{ thermal units.}$$

If, on the other hand the 1 kilo of the mixture takes up the extra heat-quantity Q_x by superheating on its way to the cylinder, then the total heat imparted to it up to admission is

$$\lambda = q + r + c_p\,(t_x - t) \text{ thermal units}$$

(for c_p see § 13, 3). In both cases the heat quantity that must be supplied to the weight G of the mixture used per stroke is

$$G\lambda \text{ thermal units.}$$

7) At the beginning of the stroke the weight G_0 of the mixture left from the last stroke, of dryness fraction x_0, remains in the clearance, the heat contained in which is therefore, by Eq. 34 p. 32 Heat per stroke.

$$G_0\,(q_0 + x_0\,\varrho_0) \text{ thermal units,}$$

so that the whole heat supplied per stroke is

$$G\lambda + G_0\,(q_0 + x_0\,\varrho_0) \text{ thermal units.}$$

8) At the end of the admission the mixture in the cylinder has passed into the condition to be distinguished by the mark 1 and is now of the dryness-fraction x_1; the heat it contains is consequently Heat at the moment of cut-off.

$$(G + G_0)\,(q_1 + x_1\,\varrho_1) \text{ thermal units,}$$

whence it follows that the mixture must have parted with the heat

$$G\lambda + G_0\,(q_0 + x_0\,\varrho_0) - (G + G_0)\,(q_1 + x_1\,\varrho_1) \text{ thermal units.}$$

9) One part of this heat $A\,L_a$ is converted into the admission-work L_a, another portion Q_a is given up by the steam to the Heat given off during admission.

*) Zeitschrift des Vereines deutscher Ingenieure 1883. P. 161.

cool cylinder walls through condensation. Neglecting the influence of eddies in the mixture which according to GRASHOF *), G. SCHMIDT, and BRAUER **) is quite unimportant, we have by the foregoing

$$A L_a + Q_a = G\lambda + G_0 (q_0 + x_0 \varrho_0) - (G + G_0) (q_1 + x_1 \varrho_1) \text{ thermal units} \quad \ldots \ldots \ldots \ldots (75)$$

Eddies neglected.

10) *The eddies in the steam* during admission are caused by the retardation of its motion, the velocity of which at admission is about 30 to 50 m per sec. Taking the velocity at an average of 40 m per sec. its "height due" is about 80 m and the kinetic energy of a kilo of steam 80 mk. Now one kilo of steam at 2 to 10 atmospheres pressure, in virtue of its external latent heat Apu (see tables pp. 28 to 31) of 42 to 46, or a mean of 44 thermal units, produces at full pressure an energy of 44×424 mk. Therefore, the kinetic energy of the eddies is to the external work of the steam as 80 is to $44 \times 424 = \frac{1}{233}$ or 0.43 %, assuming them to last till the end of the admission. As this is probably not the case, and furthermore, the work during admission cannot be so exactly recorded by the indicator that 0.43 % of it is appreciable at all, the eddies can be altogether neglected in practice.

Heat imparted to the steam during expansion.

11) **Process during expansion.** During expansion the mixture performs the work L_b, in passing from the state 1 to the state 2 while it receives back from the walls the heat Q_b through re-evaporation. Referring to the equation evolved immediately before Eq. 41 p. 37, we have, for the end of the expansion

$$A L_b - Q_b = (G + G_0)(q_1 + x_1 \varrho_1 - q_2 - x_2 \varrho_2) \text{ thermal units} \quad . \; . \; (76)$$

Heat lost during the exhaust.

12) **Process during the period of exhaust.** At the release the mixture in the cylinder contains, by Eq. 76, the heat

$$(G + G_0)(q_2 + x_2 \varrho_2) \text{ thermal units.}$$

As at the end of the exhaust the weight G_0 in the state 3 remains in the cylinder, there are left from the preceding heat-quantity

$$G_0 (q_3 + x_3 \varrho_3) \text{ thermal units.}$$

The losses are

a) the heat contained in the weight G of the condensed water of temperature t_4 thrown out by the air-pump $= G q_4$ thermal units;

b) the heat which warms the weight G_i of the condensing water from temperature t_i to t_5 in the condenser $= G_i (q_5 - q_i)$ thermal units;

*) Zeitschrift des Vereines deutscher Ingenieure. 1883. P. 173.

**) Ibid. P. 658.

c) the heat Q_i given up by the mixture to the walls of the eduction pipe, condenser, and air-pump.

13) Against this lost heat there is the heat $A L_c$ gained from the back pressure work L_c and the heat Q_c radiated from the heated walls of the cylinder towards the condenser or the atmosphere. So that we get

Heat gained during the exhaust.

$$A L + Q_c - Q_i = G q_4 + G_i (q_5 - q_i) + G_o (q_3 + x_3 \varrho_3) - (G + G_o) (q_2 + x_2 \varrho_2) \text{ thermal units} \ldots \ldots \ldots \ldots (77)$$

14) **Process during the period of compression.** By the work of compression L_d the weight G_o of the mixture is transferred from the state 3 to the state o while parting with the heat Q_d to the walls of the cylinder, whence follows

Heat given off during compression.

$$A L_d - Q_d = G_o (q_o + x_o \varrho_o - q_3 - x_3 \varrho_3) \text{ thermal units} \ldots (78)$$

15) **Permanent state of the cylinder-walls.** By the foregoing, the walls take up per stroke the heat

Heat taken up by the walls.

Q_a, from the entering steam,
Q_d, from the compressed steam,

besides which, if a jacket is used,

Q_m, from the jacket steam, and finally
Q_r, produced by the friction of the piston in the cylinder.

16) On the other hand, the wall gives off per stroke, the heat

Heat given off by the walls.

Q_b to the expanding steam,
Q_c by radiation to the condenser, and
Q_v to the surrounding air.

So that in the permanent state we must have

$$Q_a - Q_b - Q_c + Q_d + Q_m + Q_r - Q_v = O \ldots \ldots (79)$$

17) **Performance of the engine.** As the weight G of the mixture enters the cylinder with the total heat $G\lambda$ and by the foregoing, the heat-quantities Q_i, $G q_4$ and $G_i (q_5 - q_i)$ pass into the condenser, therefore the heat actually used in the engine and converted into the indicated work L is

Heat used in doing work.

$$G\lambda - G q_4 - G_i (q_5 - q_i) - Q_i = G(\lambda - q_4) - G_i (q_5 - q_i) - Q_i \text{ units of heat.}$$

Taking into account the heat supplied by the friction of the piston and by the jacket (if there is one) also the heat lost by external radiation, we have for the indicated work L of the engine, the following equation

$$A L = G(\lambda - q_4) - G_i (q_5 - q_i) - Q_i - Q_v + Q_m + Q_r \text{ units of heat } (80)$$

18) **Transformation of the principal equations.** If, as is customary, we neglect the volume of the water present compared with that of the steam, and call the volume of steam at the end of each of the different periods V and the respective density γ, (with their corresponding marks) we get

Volume of the water neglected.

$$(G+G_0)\,x_1 = V_1\,\gamma_1$$
$$(G+G_0)\,x_2 = V_2\,\gamma_2$$
$$G_0\,x_3 = V_3\,\gamma_3$$
$$G_0\,x_0 = V_0\,\gamma_0$$

Transformation of the equations.

19) Applying these designations, we have the following forms for Equations 75 to 78

$$Q_a = V_0\gamma_0\varrho_0 - V_1\gamma_1\varrho_1 + G(\lambda - q_1) - G_0(q_1 - q_0) - A\,L_a \quad (81)$$
$$Q_b = V_2\gamma_2\varrho_2 - V_1\gamma_1\varrho_1 - (G+G_0)(q_1 - q_2) + A\,L_b \quad (82)$$
$$Q_c = V_3\gamma_3\varrho_3 - V_2\gamma_2\varrho_2 - G(q_2 - q_4) - G_0(q_2 - q_3) + G_i(q_5 - q_i) - A\,L_c + Q_i \quad (83)$$
$$Q_d = V_3\gamma_3\varrho_3 - V_0\gamma_0\varrho_0 - G_0(q_0 - q_3) + A\,L_d \quad (84)$$

Example.

20) **III. Determination of the different heat-quantities.** The four heat-quantities Q_a, Q_b, Q_c, Q_d, in the above equations are the most important in judging of the efficiency of a steam-engine. To illustrate the method of determining these magnitudes GRASHOF employs the following example of ZEUNER'S, referring to HIRN'S experiment of the 27th. Aug. 1875 (mentioned in § 13. 16) with the unjacketed jet-condensing engine, using superheated steam and cutting off at 45 % of the stroke. The cylinder was 60.5 cm diar. $\times$ 170.2 cm stroke, with a clearance of 1 % of the volume swept.

Given quantities.

21) **Given quantities.** For computing the different heat-quantities, we have first the four volumes V_1, V_2, V_3, V_4 given, as they are known from the dimensions of the cylinder and the setting of the valve-gear. In HIRN'S engine (the figures for which are shewn below in brackets) they were respectively

0.2224, *0.4900*, *0.0400* and *0.0050* cbm.

From the indicator cards we have further in thermal units.

$A\,L_a\,(14.79)$, $A\,L_b\,(9.24)$, $A\,L_c\,(1.97)$, $A\,L_d\,(0.20)$, $A\,L\,(21.86)$.

We can now take, from the steam table pp. 28 to 31, the values

$q_1\,(125.00)$, $q_2\,(94.79)$, $q_3\,(52.96)$, $q_0\,(84.80)$ thermal units,
$\varrho_1\,(477.22)$, $\varrho_2\,(500.76)$, $\varrho_3\,(533.59)$, $\varrho_0\,(5.0858)$ " "
$\gamma_1\,(1.2900)$, $\gamma_2\,(0.4996)$, $\gamma_3\,(0.0945)$, $\gamma_0\,(0.3507)$ kilos,

corresponding to the indicator pressures in kilos per □ cm at the end of the respective periods, viz.

$p_1\,(2.3070)$, $p_2\,(0.8417)$, $p_3\,(0.1447)$; $p_0\,(0.5787)$.

Next $t_i\,(16.5^0)$, $t_4\,(35.26^0)$ and $t_5\,(35.26^0)$ can be observed, by which

$q_i\,(16.51)$, $q_4\,(35.30)$ and $q_5\,(35.30)$ thermal units

are given. The pressure in the boiler and the state of the steam at the end of the main steam pipe near the engine being noted, λ is then determined by help of the data contained in 6). In HIRN'S experiment p was $= 4.8075$ kg/□ cm;

$t = 150^0$ and $t_x = 223^0$, whence with $c_p = 0.48$, $\lambda = 687.29$ thermal units. The feed-water $G (0.2822)$ and condensing water $G_i (8.5983)$ per stroke in kilos can be measured. The remaining unknown quantities can be obtained as follows by experiment, estimation, or calculation.

22) **The heat Q_m given up by the jacket-steam** can be deduced from the weight of condensed water collecting in the jacket in a given time, assuming the condition of the steam (generally boiler steam), before it enters the jacket, to be known. The tables on pp. 120 and 121 give Q_m for marine engines as, on an average, about 3% of the total steam used. In a small one-cylinder engine of 30 cm diar. and 75 cm stroke working at 75 revolutions, FLIEGNER*) found the heat per stroke given up by the jacket steam (of 3.81 k per □ cm pressure) to the steam in the cylinder to be 1.618 thermal units, or in round numbers 2.5 thermal units per sec. per □ m of jacketed surface. For HIRN'S unjacketed engine of course $Q_m = 0$. **Heat imparted to the cylinder from the jacket.**

23) **The heat radiated on the outside of the cylinder Q_v in a jacketed engine** can easily be ascertained by turning the steam on to the jacket when the engine is stopped. When the cylinder itself has been by this means heated to the full temperature of the steam, the water collecting in the jacket in a certain time corresponding to a certain number of strokes is to be measured. As the cylinder is assumed to be closed, this water must be entirely due to the heat given off on the outside, through the jacket. According to the table on pp. 120 and 121, the heat Q_v is almost constant for the same marine engine under different circumstances see ("Duquesne" and "Mytho"). On an average, it may be estimated, as these experiments shew, at about 1.3 to 1.5% of the total heat supplied per stroke when the engine is developing its utmost power. **Exterior radiation from the jacket.**

24) *In an unjacketed engine* Q_v is less easily determined by experiment than in a jacketed engine, but assuming equally efficient lagging, Q_v in this case is smaller and therefore of less importance. In estimating it, we may take LONGRIDGE'S figures for Q_v as a basis, from the table on p. 97, column 7, referring to the unjacketed cylinder ends of his compound engine, the average being 0.15 thermal units per □ m of surface per stroke. For HIRN'S engine GRASHOF estimates Q_v at 0.75 thermal units per stroke corresponding to about 0.2 thermal units per □ m per stroke, which agrees very well with **Exterior radiation of an unjacketed cylinder.**

*) Schweizerische Bauzeitung. Vol. XII. Nr. 13 & 14.

LONGRIDGE'S value, when we consider that his is probably rather too low, as it only relates to the cylinder ends where the heat transmission is known by experience to be slower than in the sides.

Heat gained by piston friction.

25) **The heat of the piston friction** Q_r is deduced by GRASHOF from the experiments carried out by VOELCKERS with six different engines running empty, after he had separately determined the friction in the bearings of the flywheel shaft. It was found that the load per □cm on the piston necessary to overcome the friction of the piston, connecting-rod, guides, and valve-gear and to drive the feed-pump amounted to $\frac{0.0227}{d}$ atmospheres, d. being the diar. of the cylinder in metres. We therefore have, for HIRN'S engine

$$\frac{0.0227 \times 10333 \times \pi \times 0.605^2 \times 1.702}{4 \times 0.605} A = 0.44 \text{ thermal units}$$

half of which, or 0.22 thermal units would probably sufficiently account for the piston-friction even when the engine was loaded. In this case for HIRN'S engine Q_r would be about 0.01 AL or 1 $^0/_0$ of AL, which will serve as a basis for estimating Q_r. — LONGRIDGE in his experiments allows Q_r to be 0.05 L which, as he particularly observes, is well on the safe side. This view has recently been corroborated by a report of THURSTON*) on friction experiments with steam engines which took place in 1888 at Ithaca N. Y. The work of the piston-friction of a 20 h. p. horizontal engine was there found to be only 0.065 $^0/_0$ of its gross $IH\!P$ so that $Q_r = 0.00065\, A\, L$. In general therefore, it appears that Q_r can be safely neglected.

Heat given up to the condenser walls.

26) **The heat given up to the walls of the eduction-pipe, condenser, and air-pump** Q_i can be calculated from Eq. 80 after determining Q_m, Q_v, and Q_r. For HIRN'S engine GRASHOF takes the *average* value of $Q_i = \frac{1}{2} Q_v$ and thus determines $Q_i = 0.04$ thermal units per stroke, a very small quantity. ZEUNER entirely neglects Q_v, probably because of its smallness. GRASHOF also is chiefly desirous of determining Q_i as a means of correcting any errors of observation affecting the other terms of Eq. 80, because as G_i is of considerable magnitude (see in a later paragraph "quantity of injection and circulating water") minute errors of observation in t_i and t_5 amounting only to small

*) Paper read before the american society of mechanical engineers at the Scranton meeting. 15—19 October 1888.

fractions of a degree may lead to great differences in the value of Q_i.

27) **The heat taken up by the cylinder-walls during the period of compression** Q_d is assumed by GRASHOF to be nothing for *unjacketed* cylinders, because it is pretty certain that at the beginning of the compression the wall is parting with heat to the steam and that this process is not reversed till later. For *jacketed* cylinders Q_d is to be calculated by Eq. 84, for which G_0 is first to be determined. With regard to this, HIRN and HALLAUER assume the steam to be dry and saturated at the beginning or at the end of the compression so that x_3 or x_0 would be $= 1$ and $\therefore G_0 = V_3 \gamma_3$ or $= V_0 \gamma_0$, a comparatively small quantity, which can in many cases, be neglected. Heat given up to the cylinder walls during compression.

28) ZEUNER*) however, is of opinion that a certain quantity of water remains in the mass G_0 which should not be neglected with a high rate of compression. BRAUER**), while sharing ZEUNER'S view as to the existence of this water, is convinced that, so long as its volume can be neglected in comparison to the volume of the clearance, it has no effect upon the calculation. While the engine is working regularly, if there is any water left in the cylinder during compression it will alternately receive and give up heat like the cylinder wall, but in a much smaller degree. The weight of this water is always the same for the same position of the piston, and a knowledge of its weight is no more necessary, for the equations of the heat in the steam, than of the weight of the cylinder itself. We may therefore look upon the water contained in the mixture G_0 as part and parcel of the cylinder-wall, and assuming G_0 to designate only the vapourous portion of the mixture, we may, with HIRN and HALLAUER, regard the water as non-existent. Water residue in the compression steam.

29) HIRN'S investigations shew that in most cases the amounts of the particularly important heat-quantities Q_a, Q_b, and Q_c are considerably influenced by assuming different values for G_0. For instance, if we take $Q_d = 0$, it follows by Eq. 84 that $G_0 = 0.0416$ kilos, but if we put $x_1 = 1$ (for $x_3 = 1$ or $x_0 = 1$ we already get $x_1 > 1$), then by the corresponding equation in 18), $G_0 = 0.0047$ kilos; i. e. the former value is nine times greater than the latter, and yet both assumptions are equally permissible, or equally arbitrary. As a rule however the former assumption of HIRN'S (27) is kept to. Influence of the weight of the compression steam.

*) Civilingenieur 1881. Calorimetrische Untersuchung der Dampfmaschinen.

**) Zeitschrift des Vereines deutscher Ingenieure 1883. P. 658.

Exchange of heat during admission and expansion.

30) **The heat Q_a given up by the admission steam to the wall, and the heat Q_b given back by the wall to the expanding steam** can now be determined by Eq. 81 and Eq. 82. For HIRN'S engine we have

for $G_0 = 0.0416$, $x_1 = 0.886$ and $x_2 = 0.756$

$Q_a = 6.19$ thermal units and $Q_b = -14.87$ thermal units,

and for $G_0 = 0.0047$, $x_1 = 1$ and $x_2 = 0.853$

$Q_a = 7.68$ thermal units and $Q_b = -13.75$ thermal units.

Here it is worthy of note that Q_b is negative, i. e. that the wall gives up no heat to the expanding steam but that more heat is withdrawn from the latter during expansion than during admission. This circumstance arises from the use of superheated steam in a condensing engine without a jacket (see § 13, 16). At the same time, the negative value of Q_b does not necessarily signify that there is not some heat given up by the wall to the steam towards the end of the expansion, as on the other hand a positive value of Q_b does not exclude the transition of heat from the steam to the wall which probably always takes place at the beginning of the expansion. In HIRN'S engine when working with superheated steam there was no re-evaporation on account of the negative value of Q_b; on the other hand the indicator cards shewed a not inconsiderable diminution of the visible quantity of steam towards the end of the stroke. This fact, together with the loss of heat determined by GRASHOF of $Q_a + Q_b = 21.06$ to 21.43 thermal units for 0.45 cut-off with a total of $A L = 21.86$ usefully converted thermal units, cannot well be reconciled with HIRN'S statement that his engine, at 0.25 cut-off, and 27 revolutions with steam superheated to 230° at 4.5 atmos. pressure, only shewed 6.5 % of internal condensation, which rose to 30.4 % when the engine was working under the same conditions but without superheating.

English's experiments.

31) The latest experiments undertaken with the object of measuring the nett amount of initial condensation in the cylinder, i. e. of determining the difference $Q_a - Q_b$, are those of ENGLISH *). KNOKE **) reports upon them as follows: the small portable engine used in the trials had a jacketed cylinder of 25.4 cm diar. and 35.6 cm stroke. The weight of steam per stroke was so small and the admission of it so direct as almost to preclude the possibility of priming or condensation on the way to the cylinder. The piston was fixed at the

*) Engineering 1887. II. P. 386.

**) Zeitschrift des Vereines deutscher Ingenieure. 1888. P. 316.

back centre and the connecting-rod removed. The inside of the cylinder round the piston-rod was blocked with pieces of iron and wood, as well as the front steam port, the entrance of which was closed with a carefully-fitted brass plate. The crankshaft, excentric, and slide were driven by another engine. The boiler pressure was kept steady and the stop-valve wide open. Consequently, at every revolution boiler steam entered the back clearance and afterwards escaped by the exhaust. The slide cut-off at 0.7 of the stroke. Each experiment lasted one hour, the revolutions were noted by means of a counter and three indicator cards were taken at certain intervals. The exhaust steam was measured with a surface condenser. The deficiency between the weight of the exhaust steam and that of the dry steam required to fill the clearance of 0.99 litres then gave the difference between the weight of steam condensed on the surface of the clearance space of 0.1858 □ m, and the weight re-evaporated. Altogether 64 experiments were carried out, 35 with the condenser and 29 exhausting into the air. The boiler pressures used were 3.16, 2.11, 1.41, 0.7 kilos per □ cm and the corresponding revolutions per minute 130, 100, 70, and 50. The results of these experiments are given in the following table, the figures in which are mean values, so that they do *not* agree together.

Table shewing the initial condensation in jacketed cylinders.

	Clearance	Revs. per sec.	Density of the steam		Thermal units in 1 kilo of		Feed water per rev.	Initial condensation in thermal units	
			admission	release	steam	water		total per rev.	per □ m surface at 1 rev. per sec.
	V_0 cbm	N	γ_1	γ_0	λ_1	q_1	kilos	C	$\frac{C\sqrt{N}}{O_0}$
1	2	3	4	5	6	7	8	9	10
Noncondensing	0,00099	1,486	1,746	0,652	647,22	133,89	0,00617	2,598	16,44
Condensing	0,00102	1,459	1,682	0,279	646,67	132,78	0,00640	2,523	15,60

The figures in columns 9 and 10 of this table seem to me scarcely to deserve complete confidence. It is remarkable that the internal condensation should be given as 5.4 % greater when the engine was making 89 revolutions per min. with a range of temperature of 33° and exhausting into the air, than at 87.5 revolutions with 55° range of temperature, working as a condensing engine. Further, these values must be called very large compared with those found by LONGRIDGE for his

compound engine (see table p. 97, col. 6) and he therefore thinks the piston of ENGLISH's engine cannot have been tight. Finally, UNWIN and others consider its weight of steam used very high, so that it cannot be considered a good engine, if only on account of its large clearance. Nevertheless the results of ENGLISH's experiments deserve, among the very scanty literature on the subject, to be highly appreciated, even if they do nothing more than demonstrate the extreme difficulty of deducing incontestably correct numerical values from such trials. There is no explanation for this increased internal condensation in the non-condensing engine with its higher exhaust temperature compared with the condensing engine; although the frequent observation of about equal internal condensation in both non-condensing and condensing engines, under the same conditions, is probably due to the fact that the gain from this source is much smaller in the former than in the latter on account of the higher pressure and temperature under which the re-evaporation takes place.

Determination of C.

32) The quantity C in the foregoing table is determined as follows.
Let γ_1 = the density of the steam at cut-off,
t_1 = „ temperature of the steam at cut-off,
γ_0 = „ density of the exhaust steam at beginning of compression,
t_0 = „ temperature of the exhaust steam at beginning of compression,
G_0 = „ weight of water in kilos collected in the condenser per revolution,
V_0 = „ volume of the clearance in cbm.

There are condensed and not re-evaporated per revolution

$$G = G^1 - V_0\,(\gamma_1 - \gamma_0) \text{ kilos of steam.}$$

Every kilo of this steam at temperature t_1 required λ_1 thermal units to produce it and if q_1 is the corresponding "heat of the liquid", every kilo of steam condensed in the cylinder has lost the heat $\lambda_1 - q_1 = r_1$. Therefore there are withdrawn from the steam in the clearance per revolution between the opening and closing of the port

$$C = G\,r = [G' - V_0\,(\gamma_1 - \gamma_0)]\,(\lambda_1 - q_1) \text{ thermal units} \quad . \; . \; (85)$$

Inferences from these trials.

33) It may be noted that ENGLISH's experiments point to the same law as KIRSCH's*) theoretical investigations (see § 15, 16 and § 16, 8) viz. *that the amount of initial condensation minus the quantity re-evaporated*, or in other words, *the heat taken up*

*) KIRSCH. Die Bewegung der Wärme in den Cylinderwandungen der Dampfmaschinen. Leipzig 1886. P. 10.

by the cylinder walls is, ceteris paribus inversely proportional to the square-root of the number of revolutions. With regard to DELAFOND's observation from his experiments (§ 16, 23) that the influence of the revolutions on the internal condensation was much greater, indeed that the latter was inversely proportional to the number of revolutions and not to the square-root of that number, LÜDERS*) is of opinion that this arose from the circumstance that with the greater speed the gains of heat due to piston-friction and re-evaporation increase, while the loss from external radiation is unaltered, as already referred to in 23). Perhaps also the more forcible exhaust corresponding to the higher speed carries over mechanically a portion of the water contained in the steam which therefore does not require to be re-evaporated at the expense of the heat of the cylinder walls. This view is to a certain extent corroborated by ESCHER**), who found that the internal condensation diminished faster than the square-root of the revolutions increased. But it follows from § 15, 18 to 24, that this loss cannot be simply in the inverse ratio of the revolutions. GRASHOF***) besides, demonstrates particularly that from the moment the entering steam comes in contact with the cooler wall, the gradient of temperature between the innermost and the succeeding elementary layer of the wall falls steadily from an infinitely high value, as shewn by Figs. 2 and 3, Plate 2. So that as the flow of heat into the wall and therefore the accumulation of water on its internal surface are getting steadily slower, the internal condensation must decrease in a lesser ratio than that of the increase of the revolutions. Further experiment is therefore required in order to confirm the correctness of the above law as theoretically deduced by KIRSCH.

34) *The magnitude of the initial transmission of heat* from the steam to the wall, and therefore of the internal condensation, depends however (besides the revolutions) upon the ratio of cut-off, to the square-root of which it appears, by the American experiments § 16, 8, to be likewise inversely proportional. Further, it depends upon the size of the cylinder (§ 16, 33), the range of temperature (§ 16, 36), and the manner in which the cylinder is heated or cooled externally (§ 16, 14). Finally, the degree of moisture (or in other words, the dryness-fraction) of the steam materially influences the amount of internal condensation, as the follow- **Magnitude of the initial transmission of heat.**

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*) Zeitschrift des Vereines deutscher Ingenieure. 1889. P. 356.

**) ESCHER. Versuche über die Dampfverluste in der Dampfmaschine. Civilingenieur. Vol. XXVII.

***) F. GRASHOF. Theoretische Maschinenlehre. Vol. III. P. 557. Hamburg 1888.

ing considerations shew. DONKIN*) has recently observed, by means of a glass vessel connected to the cylinder, that part of the watery precipitate present at the release is dispersed in the form of minute particles by the sudden fall in the pressure and carried out of the cylinder with the steam. It is also surmised that the water formed during expansion in consequence of the steam performing work (§ 11, 5) does not collect on the cylinder walls, but remains floating in the steam and is likewise carried away by the exhaust. In this way the re-evaporation by the cylinder walls of the water precipitated from the entering steam is considerably facilitated, there may even remain in them a small excess of heat when perfectly dry steam is used, as the re-evaporation requires less heat than originally passed into the wall during the condensation. This excess, which with dry steam raises the mean temperature of the wall and therefore reduces the internal condensation, may serve, up to a certain degree, to evaporate any moisture the steam contains and thus again, by reducing the temperature of the walls, to increase the internal condensation. If, however, the steam is so loaded with water that the excess of heat in the walls no longer suffices to evaporate it, liquid water will collect in the cylinder, but will be mechanically carried off at the release, as above described. So long as this is completely effected, the moisture of the steam will not give rise to any particular losses of heat, but if the degree of moisture exceeds this limit, the amount of internal condensation increases and is evidenced by heavy re-evaporation which of course means losses of heat as already shewn in § 15, 20. The injurious effect of wet steam upon the working of engines is further gone into under superheaters. The quantity of heat given up to 1 □m of the wall surface at 60 revolutions per minute serves as a measure of the magnitude of the internal condensation. LÜDERS**) calculates it for

ENGLISH's experiments	at 26	thermal units,
a very good engine, cutting off at 25 % (actual)	„ 26.5	„ „ ,
ISHERWOOD's trials of the Paddle Sloop "Michigan" (see p. 80)	28.3 to 31.5	„ „ ,
EMERY's trials of the Steamer "Dexter" (table p. 86, col. 8)	21.7	„ „ ,
EMERY's experiment with the Steamer "Dallas" (table p. 86, col. 4)	15.0	„ „ ,

*) The Engineer. 1889. I. P. 163.

**) Zeitschrift des Vereines deutscher Ingenieure. 1889. P. 570.

ISHERWOOD's trial of a small one-horse engine 28 thermal units, DELAFOND's experiments shewed a smaller amount than the mean of the above.

35) ENGLISH propounds the following empirical formula, for the heat lost by initial internal condensation and not recovered in re-evaporation, based on his own experiments and those of EMERY treated of in § 16, 18 to 20. In this formula — **Formula for the heat lost by internal condensation.**

λ_1' = the sum of the units of heat contained in 1 kilo of the mixture of steam and water at any point of the stroke and corresponding to the work done (up to that point),

O_0 = the surface of the clearance in □ m.

O' = „ „ „ „ cylinder wall up to the particular point of the stroke.

He then gives, for cylinders *with* jackets

$$G'(\lambda - \lambda_1') = 1{,}381 \frac{\gamma_1 O_0}{\sqrt{N}} \left(1 - \frac{\gamma_0 + 0{,}96}{\gamma_1} \cdot \frac{O'}{O_0}\right) \text{ thermal units} \quad \ldots . (86)$$

and for cylinders *without* jackets

$$G'(\lambda - \lambda_1') = 1{,}841 \frac{\gamma_1 O_0}{\sqrt{N}} \left(1 - \frac{\gamma_0 + 0{,}96}{\gamma_1} \cdot \frac{O'}{O_0}\right) \quad „ \quad „ \quad \ldots . (87)$$

36) **The heat radiated into the condenser &c. or into the atmosphere,** Q_c is calculated from Eq. 83. Taking HIRN's engine as an example, we get — **Heat radiated into the condenser &c.**

for $G_0 = 0{,}0416$; $Q_c = 20{,}53$ thermal units,

and for $G_0 = 0{,}0047$; $Q_c = 22{,}08$ „ „ .

The difference in the assumed values of G_0 here causes a variation in those of Q_c of 1.55 units or about 7 % of AL (21.86 units).

37) As Q_c forms the greatest of all the losses of heat, HIRN and HALLAUER paid particular attention to it in their experiments, in order to form a conclusion as to the efficiency of the engine by comparing this loss with the amount of heat converted into useful work. The annexed table contains the results of some calorimetric researches carried out by Engineer WIDMANN of the French Navy with different engines on the occasions of their official trial trips in 1879. These results were compiled by HALLAUER*). The heat-quantity Q_c varies in these trials from 0.5 to 12.1 % of the total heat supplied in the Woolf's engine, and in the compound engines from 6.7 — **Table of the French engines.**

*) Bulletin de la société industrielle de Mulhouse. 1880. P. 212.

to 21.9 %, a point to which further reference will be made later on.

Efficiency.

38) The efficiency-ratio of a steam engine, which is the principal thing to be determined from the heat trials, cannot be estimated with certainly from Q_c, the heat radiated into the condenser, because this value varies according to that assumed for G_0, as has been shewn. Differing from HIRN and HALLAUER, GRASHOF proves very correctly that the efficiency ratio can only be determined by comparing the useful heat AL with the heat expended in forming the steam plus that used in heating the cylinder Q_m. The heat used for the formation of the steam, assuming the water to be returned to the boiler at the temperature t_4 at which it is thrown out by the air-pump, is $G(\lambda - q_4)$ and therefore the efficiency-ratio

$$\eta = \frac{AL}{G(\lambda - q_4) + Q_m} \quad \ldots\ldots\ldots\ldots \quad (88)$$

For HIRN's engine, with any assumed value of G_0, and in spite of the correspondingly different values of Q_c, this ratio is always $\eta = 0{,}119$. For the marine engines referred to in the foregoing table the mean value of η_c is 0.12, the extremes being about 0.11 and 0.14. KENNEDY's 17 hours' trial of the triple-expansion engines of the S. S. "Meteor" in June 1888*) gave $\eta_c = 0.162$.

Absolute magnitude of the thermal efficiency.

39) This thermal efficiency of marine engines appears at first sight extraordinarily low. It must however be borne in mind that by Eq. 21 (p. 20) even in a perfect cycle it is only possible, out of a given heat quantity Q, to convert the portion

$$AL = \frac{Q}{T}(T - T_1)$$

into work, so that the thermal efficiency of this cycle can only be

$$\eta_c = \frac{Q(T - T_1)}{LT} = \frac{T - T_1}{T}, \quad \ldots\ldots\ldots\ldots \quad (89)$$

which value becomes a maximum, i. e. equal to unity for $T_1 = 0$. In a steam-engine (without superheating) the initial absolute temperature T depends upon the saturation temperature t of the boiler steam, so that T can be at most $273 + t$. On the other hand the terminal absolute temperature never sinks to the absolute zero $T_1 = 0$, but is limited by the back pressure which, in condensing engines, is at least 0.1 atmos. (say 1.5 ℔s) so that $t_1 = 46{,}2^0$ or $T_1 = 273 + 46{,}2 = 319{,}2^0$.

*) Proceedings of the institution of mechanical engineers. June 1889.

We thus get, for instance, with a low-pressure condensing engine, working at 2 atmos. (say 30 ℔s), that is where $t_1 = 133{,}9^0$ or $T_1 = 273 + 133{,}9 = 406{,}9$, a theoretical thermal efficiency of

$$\eta_c = \frac{406{,}9 - 319{,}2}{406{,}9} = 0{,}216.$$

ZEUNER *) has calculated the effect of the deviation of the process in the cylinder from a perfect cycle and of the difference between the changes of state of steam and those of a perfect gas. He obtains the following results:

Working pressure in atmospheres	2	4	9
Thermal efficiency of the perfect cycle .	0,216	0,249	0,296
" " " " steam engine .	0,201	0,229	0,287
Difference between them as a percentage of the former	6,94	8,03	9,80.

40) The French marine engines, the caloric efficiency of which was found by the experiments to be 0.12 mean, all worked at about 4 atmos. (60 ℔s) so that their highest attainable theoretical efficiency was 0.229. In reality therefore these engines converted as a mean $\frac{0{,}12 \times 100}{0{,}229} = 52{,}5\,^0/_0$, and in the most favourable case with $\eta_c = 0{,}14$, about 61 $^0/_0$ of the total heat theoretically convertible into work; the "Meteor's" percentage was 62 (compare § 18, 23). This percentage becomes higher in each case if in calculating η_c from Eq. 89 the corresponding initial temperature of the steam in the cylinder and the temperature of the exhaust are substituted, as the theoretical value of η_c then becomes less than 0.229.

Relative magnitude of the thermal efficiency.

41) Assuming that a reversible cycle of a perfect gas takes place in the steam-engine, the initial temperature of which equals the saturation temperature of the boiler steam and its terminal absolute temperature $T_1 = 313$ corresponding to a condenser temperature of about 40^0, we get as the highest possible theoretical thermal efficiency for

Value of η_c for estimating.

steam of	2	4	6	8	10	12	14	16	18	20	W. P. in atmos.
η_c =	0,229	0,260	0,283	0,300	0,313	0,324	0,333	0,342	0,350	0,357	

In practice it is sufficiently exact to deduct 7 to 10 $^0/_0$ from these values (see end of 39), in estimating the highest thermal efficiency of an engine at the usual working pressures of from 2 to 10 atmos.

*) G. ZEUNER. Grundzüge der mechanischen Wärmetheorie. Leipzig 1877. PP. 464 & 465.

Table of the heat trial

	1ste Class "Du	
1	2	3
Type of engine	3 coupled horizontal engines with 3	
IHP	8490,000	7200,000
Ratio of volume of *HP* cylinder to that of *LP* cylinder	0,519	0,519
" " cut-off in *HP* cylinder	0,725	0,6[illegible]
Gross ratio of cut-off	0,376	0,3[illegible]
Revolutions per minute	80,830	76,6[illegible]
Weight[5]) of mixture of steam & water in the *HP* cylinder or cylinders in kilos	9,273	8,1[illegible]
Calculated weight of dry steam present in this mixture at the moment of cut-off in kilos	9,255	8,1[illegible]
Weight of water contained in the mixture in kilos	0,018	0,01[illegible]
Percentage of water to whole weight of mixture admitted	0,200	0,00[illegible]
"Heat of the steam" at moment of cut-off, T_0 in thermal units	5597,000	4903,00[illegible]
Calculated weight of dry steam in the *HP* cylr. or cylrs. at end of stroke in kilos	9,276	8,361
Weight of water contained in the mixture at the same moment in kilos	0,003	— 0,240
Percentage of weight of water to whole weight of mixture	steam dry	steam superheated
"Heat of the steam" at the end of the stroke in the *HP* cyl. or cyls. T_2 in thermal units	5588,000	4998,00[illegible]
Weight of mixture of steam & water in the *LP* cylr. or cylrs. in kilos	8,646	7,67[illegible]
Calculated weight of dry steam contained in the *LP* cyl. or cyls. at end of stroke in kilos	8,031	7,00[illegible]
Weight of water contained in the mixture in kilos	0,615	0,66[illegible]
Percentage of water to whole weight of mixture admitted to *LP* cylinder	7,100	8,60[illegible]
"Heat of the steam" at end of stroke in the *LP* cylr. or cylrs. T_1 in thermal units	4849,000	4237,00[illegible]
Difference between the initial & terminal "heat of the steam" ($T_0 - T_1$) in thermal units	+ 748,000	+ 666,00[illegible]
" " heat of the steam at the beginning and end of the compression in thermal units	— 292,000	— 210,00[illegible]
Heat supplied by condensation of the jacket steam, Q_m in thermal units	+ 84,000	+ 138,00[illegible]
" " " " " " steam in the *HP* cylr. or cylrs. in thermal units	— 18,000	0,000
" used in the total work of compression in thermal units	— 429,000	— 390,000
" lost by radiation from the cylinders Q_v " " "	— 66,000	— 66,000
" radiated into the condenser Q_c " " "	27,000	138,00[illegible]
Percentage of the latter heat to the total heat supplied to the engine	0,500	2,[illegible]
Heat supplied to the cylinders in the dry steam in thermal units	5230,000	4596,00[illegible]
" " " " " by the steam condensed in the jackets in thermal units	84,000	138,00[illegible]
" " in the water carried over with the steam in thermal units	23,000	26,00[illegible]
Total heat supplied to the engine	5337,000	4760,00[illegible]
Weight of dry steam per stroke in kilos	8,225	7,3[illegible]
" " " " " *IHP* per hour	9,405	9,[illegible]

f French marine engines.

ruiser uesne"			Transport „Vienne"	Dispatch boat „Cigale"	Transport „Mytho"				Dispatch boat „Nièvre" [1])
4	5	6	7	8	9	10	11	12	13
ndem compound [2]) anks at 120°			2 cylinders inverted d. a. compound	2 cylinders inverted d. a. compound	3 cylinders inverted direct-acting compound				3 cylinders inverted d. a. compound
6360,000	3900,000	1665,000	690,000	205,000	[3]) 2590,000	2200,000	1350,000	[4]) 590,000	740,000
0,519	0,519	0,519	0,317	0,309	0,282	0,282	0,282	0,282	0,380
0,550	0,225	0,100	0,660	0,750	0,750	0,690	0,601	0,601	0,499
0,285	0,126	0,052	0,209	0,232	0,212	0,194	0,169	0,169	0,189
73,000	62,490	46,550	75,000	90,000	66,000	73,160	61,400	44,800	93,000
7,113	4,746	3,318	0,6913	0,1697	2,913	2,305	1,699	1,134	0,6201
6,651	3,609	1,671	0,6221	0,1605	2,569	1,927	1,320	0,827	0,4725
0,462	1,137	1,647	0,0692	0,0092	0,344	0,378	0,379	0,307	0,1476
6,500	24,000	49,500	10,0000	5,4000	11,800	16,400	22,300	27,100	23,8000
4076,000	2337,000	1214,000	358,8000	99,1000	1608,000	1210,000	843,000	529,000	308,4000
6,981	4,488	2,637	0,6592	0,1645	2,561	1,954	1,379	0,881	0,5343
0,132	0,258	0,681	0,0321	0,0052	0,352	0,351	0,320	0,253	0,0858
1,800	5,400	20,500	4,6000	3,1000	12,100	15,200	18,800	22,400	13,8000
4206,000	2705,000	1625,000	401,2000	100,4000	1593,000	1219,000	860,000	550,000	333,0000
6,705	4,565	3,303	0,6633	0,1600	2,800	2,225	1,565	1,080	0,6057
5,748	3,792	2,655	0,5648	0,1364	2,204	1,652	1,102	0,751	0,4082
0,957	0,864	0,648	0,0985	0,0236	0,596	0,573	0,453	0,329	0,1975
14,400	18,600	19,500	14,8000	14,7000	21,000	25,700	22,500	30,500	32,6000
3501,000	2282,000	1611,000	343,6000	83,6000	1361,000	1024,000	682,000	461,000	259,7000
- 575,000	+ 55,000	− 397,000	+ 42,2000	+ 15,5000	+ 247,000	+ 186,000	+ 161,000	+ 68,000	+ 48,7000
- 225,000	− 39,000	− 1,000	− 13,1000	− 4,6000	− 51,000	− 35,000	− 64,000	− 25,000	− 6,3000
- 138,000	+ 114,000	+ 98,000	+ 20,500	+ 4,200	+ 46,000	+ 39,000	+ 39,000	+ 32,000	+ 5,9000
- 150,000	+ 525,000	+ 821,000	+ 24,900	+ 2,100	+ 146,000	+ 172,000	+ 180,000	+ 152,000	+ 65,4000
345,000	− 317,000	− 217,000	− 39,700	− 8,200	− 145,000	− 118,000	− 88,000	− 56,000	− 33,7000
– 66,000	− 60,000	− 60,000	− 6,000	− 1,500	− 23,000	− 23,000	− 23,000	− 22,000	− 9,0000
227,000	278,000	244,000	28,800	7,500	220,000	221,000	205,000	149,000	71,0000
5,600	9,800	12,100	6,700	7,200	12,300	15,800	20,600	21,900	18,7000
4032,000	2694,000	1914,000	409,600	99,100	1729,000	1349,000	946,000	646,000	369,8000
138,000	114,000	98,000	20,500	4,200	46,000	39,000	39,000	32,000	5,9000
23,000	15,000	8,000	3,000	0,700	9,000	7,000	5,000	3,000	2,8000
4193,000	2823,000	2020,000	433,100	104,000	1784,000	1395,000	990,000	681,000	378,5000
6,473	4,353	3,117	0,6645	0,1591	2,731	2,137	1,515	1,042	0,5797
8,907	8,335	10,448	8,6750	8,3900	8,350	8,510	8,263	9,504	8,7100

Remarks. (14)

[1]) All the engines had jackets only the cylindrical surface of the "Nièvres" *LP* cylinders were unjacketed.

[2]) The three compound engines of which the "Duquesne's" engine was composed, each consisted of a horizontal tandem engine with a receiver between the cylinders.

[3]) Except in the trial in col. 9 the ships were under way. This experiment was made to determine some qualities of a particular kind of propeller for which purpose it was necessary to work the engines at moorings.

[4]) In all the trials the main stopvalves were wide open, except in that in col. 12 in which the steam was wire-drawn so as to produce a difference of 0,93 kilos per □ cm between the boiler and admission pressures.

[5]) All weights and heat-quantities, except those in the bottom line, refer to one stroke.

Value of heat trials.

42) Heat trials alone can assist us to obtain a clear view of what takes place in the cylinder and to further develop the theory of the steam engine. But as the active mixture of steam and water is in contact with a mass of metal many times heavier than itself and is perpetually giving up to and receiving from it heat quantities which may under certain circumstances be as great as the heat-equivalent AL of the whole indicated work, it is necessary to follow not only the changes of state of the steam, but the changes of temperature of the cylinder. The difficulty of the latter investigation is greatly increased by the fact that a knowledge of the mean temperature at any particular part of the cylinder is not sufficient, but that we must have the momentary temperature of every point in the cylinder for every part of the stroke.

Concluding remarks.

43) Not until we are in a position to study the results of such thermal investigations as these, in combination with very exact indicator cards, shall we be able to formulate the expansion and compression curves in such a practical shape that only the respective coefficients will require to be adapted to any particular circumstances. Till then, purely calorimetric investigations will remain of less practical importance than those at present universally carried out with the indicator.

§ 18.

Indicator trials of marine engines.

Object.

1) **I. Object of investigations with the Indicator.** As a rule indicator cards are taken off new engines during a trial-trip. The arrangement of the indicator, the method of using it, and its manifold applicability, are described at length in a later section. Formerly indicator cards were scarcely used for anything but determining the *IHP* and judging of the *distribution* of the steam; but since the introduction of the compound engine they have assumed further importance as a means of investigating the *economy* with which the steam is used.

Economy of steam.

2) One of the first points to be investigated is the ratio of the external work which should theoretically be produced by the weight of steam admitted to the cylinder per stroke to the work actually performed by it as shewn by the indicator. It is also to be seen how the expansion and compression lines of the cards compare with the theoretical ones expected and we can then investigate from any deviations between these

lines the causes of the difference between the indicated and the theoretical performance. Finally, the amount of internal condensation, as well as the dryness-fraction of the steam in the cylinders is to be determined.

3) The only proper standard of economy in the use of the steam is the theoretical performance due to a weight of steam equal to the weight of the feedwater per stroke. It is only recently that *this* has been taken as the basis of the investigation or at any rate it was only very rarely done formerly, which is due partly to the different ideals up to which designers have tried to work, and partly to the difficulty or even impossibility of obtaining all the necessary data for the construction of the exact theoretical diagram. **Standard of economy.**

4) In the following, after a short reference to the "indicator card" and the theoretical "diagram of energy", the various methods of conducting the investigations based upon the indicator trials will be described. **Division of the subject.**

5) **II. The diagram of energy traced** by the indicator is composed of **The indicator card.**

a) The line of lead
b) " " " admission
c) " " " expansion
d) " " " exhaust
e) " " " back pressure
f) " " " compression.

6) a. *The line of lead* (Fig. 6, Plate 2) rises, from 0, the point at which the slide, shortly before the "centre", opens the port, in most cases nearly vertically up to 1; because at this period the piston is nearly at rest while the slide is travelling pretty fast, so that the port is rapidly opened. The line of lead 01 shews how the pressure of the steam suddenly increases from what it was in front of the piston (at the end of the compression) to what it is in the steam-chest. **The line of lead.**

7) b. *The admission line* from 1 to 2 is next described during the influx of the steam through the open port. If the valve gear is well designed this line must be nearly parallel to the atmospheric line, that is, it should not tend much to approach this line. If this takes place it always shews that the admission is not well arranged. **The admission line.**

8) c. *The expansion line* from 2 to 3 shews the variations of the pressure after the cut-off. With the ordinary slide-valve gear the cut-off never takes place quite suddenly, but gradually, and this diminution of the opening is marked by a more or less pronounced rounding-off of the line, which often makes it **Expansion line.**

difficult to determine exactly the point 2, called *the point of cut-off*. Especially when the steam is throttled, we get a very gradual rounding of the curve, as shewn in the right-hand diagram, Fig. 5, Plate 1. The practical rule is that *the point at which the slide completely closes the port is to be regarded as the point of cut-off*. According to HRABAK*) it is to be placed (Fig. 5, Plate 1) in the intersection of the tangents to the admission and expansion lines. But this process gives a *theoretical cut-off*, which does not coincide with the actual one. For instance, in the diagram under consideration the actual cut-off takes place at 0.3 H., i. e. at 0.3 of the stroke, whereas the theoretical admission ceases at 0.25 H. The exact determination of the point of cut-off is of particular importance in certain investigations with the indicator (see 45). — The theoretical lines which are used for representing the expansion line of the indicator-diagram are described in 22).

The exhaust line.

9) d. *The exhaust line* joins the expansion line, but in the form of a curve of greater declivity. The decrease of pressure during the beginning of the exhaust, from 0.85 to 0.95 of the stroke is as a rule more noticeable in condensing than in non-condensing engines. Point 3, *the release*, at which the exhaust port begins to open is also sometimes difficult to distinguish on account of the slow motion of the slide; under ordinary circumstances we may always assume it to be the point where the expansion line begins to droop visibly. The release is a point which it is frequently of importance to determine carefully, because the indicator card shews the greatest weight of steam at the end of the period of expansion, where re-evaporation has commenced. For this reason the point 3 is not only of interest in indicator experiments (see 44) generally, but is particularly so in making use of WARRINGTON's calculation of the steam used (see further on under "steam used") and RYDER's method of determining the most economical speed of ship according to the regulations of the British Navy (see further on under "trial trips").

Back-pressure line.

10) e. *The back pressure line* from 4 to 5 shews the pressure the piston has to overcome during its return stroke. In non-condensing engines it coincides pretty nearly with the atmospheric line, and in condensing engines it is the nearer to the zero-line the more perfect the action of the condenser is. In the region of point 4 the back pressure line is at its greatest dis-

*) J. HRABAK. Hilfsbuch für Dampfmaschinen-Techniker. Berlin 1883. Einleitung. P. XXVIII.

tance from the zero-line which is due to the resistance of the exhaust passages.

11) f. *The compression line* from 5 to o illustrates the increase of pressure before the piston after the closing of the exhaust-port. This line is longer or shorter according to the character of the valve-gear. In engines with single slides and early cut-off it may take up the greater part of the bottom of the card, whereas with a very late cut-off, it sometimes shrinks to a scarcely perceptible rounding-off of the corner of the diagram at o. Point 5 — *the compression point*, at which the exhaust-port is completely closed can also often not be recognized as accurately as might be wished, because the throttling of the steam, by the slow cutting off of the slide, raises the back-pressure before the exhaust is completely closed. Compression line.

12) Here it may be mentioned that according to investigations of KAŠ*), the compression line of the diagram coincides most closely with an adiabatic of the form Equation of the compression-line.

$pv^{0.9}$ = const. for unjacketed condensing engines,
$pv^{1.0}$ = „ „ „ non-condensing engines,
$pv^{1.1}$ = „ „ jacketed engines,
$pv^{1.2}$ = „ „ „ „ , working with dry steam.

SCHRÖTER**) confirms the correctness of the last index for the compression line of engines using nearly dry steam, but particularly for the *LP* cylinder of compound engines, for which he applies it. In cases, therefore, where the question of the compression is important (see further on under 51) it is advisable to construct the theoretical compression line by the method described in § 6, 26, employing the index 1.2.

13) Indicator cards which are intended to be combined for the purpose of investigating the working of an engine, are never to be taken separately but always as the *mean of the top and bottom cards.* These afford a much better insight into the engine's performance. Whenever, later on, the combination of the cards is referred to, such *mean cards* are always understood to be used. Mean diagram.

14) **III. The theoretical diagram of energy.** The data necessary for getting ont a theoretical diagram of energy are Theoretical diagram of energy.

a) the pressure of the steam,
b) the terminal volume,
c) the initial volume, and
d) the law of the change of state of the steam expanding in the cylinder.

*) J. HRABAK. Hilfsbuch für Dampfmaschinentechniker. Berlin 1883. Theoretische Beilage. P. 58.

**) Zeitschrift des Vereines deutscher Ingenieure. 1884. P. 192.

Pressure of steam.

15) a. **The pressure** is read off as the working pressure on the boiler steam-gauge and forms the initial ordinate AC vertical to the atmospheric line of the theoretical diagram (Plate 2, Fig. 6), in which it is assumed that the condensation is perfect, so that there is an absolute vacuum or zero backpressure in front of the piston. The ordinate of the initial pressure is therefore to be produced below the atmospheric line till it cuts the line parallel to it, the zero line, representing a perfect vacuum. The vertical distance of the zero-line AB from the atmospheric line must of course be set off on the same scale as is used for the pressures, which is best taken so as to correspond with one of the usual indicator springs. These are (on the Continent)

	50	25	20	15	12	10	9	8	7	6	6	5	4 mm long
for a working pressure of	1	2	3	4	5	6	7	8	9	10	11	12	15 kilos per □cm.

Here we have taken for a working pressure of 4 kilos per □cm (say 60 ℔s per □") 15 mm = 1 kilo per □cm and therefore the vertical distance of the zero-line AB from the atmospheric line = 15.5 mm, as 1 "old" atmosphere = 1.0334 kilos per □cm, $= 1.0334 \times 15 = 15.5$.

Exact determination of the zero-line.

16) For very accurate scientific investigations, it is necessary in fixing the position of the zero-line, not only to carefully compare the indicator-spring with the scale, of which more will be said later on under Indicators, but also to take the height of the barometer. Assuming the scale to be correct (here 15 mm = 1 kilo per □cm), if, for instance, the barometer-reading at the time of taking the cards is 73.54 cm, the atmospheric pressure is $1.0334 \frac{73.54}{76.00}$ kilos per □cm and the distance of the zero-line from the atmospheric line is, in this case $\frac{1.0334 \times 73.54}{76} \times 15 = 15$ mm, as the fraction is just equal to unity.

Terminal volume of the steam.

17) b. **The terminal volume of the steam** is composed of the "volume swept" by the piston during the stroke plus the volume of the clearance.

"Volume swept."

18) *The "volume swept"* is mentioned first, because the other volumes to be referred to will generally be expressed as percentages of it. As the successive volumes swept by the piston (equal to the product of the effective area of the piston [see the § on Horse-power] and the stroke) are always proportional to the sums of the strokes, because the piston area is constant, — the volumes in the proposed diagram may be

represented by the lengths of the strokes. For the sake of simplicity and convenience, it is advisable whenever it can be arranged to take the original indicator cards 100 mm long. We will therefore make the abscissa AB, which in the theoretical diagram represents the volume swept also $= 100$ mm. The diagram Fig. 6, Plate 2 was taken from an engine of 450 mm stroke, so that 1 mm in the length of this diagram represents 4.5 mm of stroke.

19) *The volume of the clearance*, or the space between the piston and cylinder end when the engine is on the top or bottom plus the contents of the steam-port, can seldom be closely calculated because of the irregular form of the latter. The easiest way to measure it is to turn the engine to either "centre", make the piston watertight with tallow or wax, and fill up the clearance space with water by means of a vessel whose contents are accurately known. This process must be gone through for each end of the cylinder, because the clearances of marine engines are seldom both alike. The mean of the two volumes is then to be taken. In the present case (Fig. 6, Plate 2), the diameter of the piston being 700 mm the mean of the two clearances amounted to 14.7 litres of water, so that it was 8.5 % of the volume swept viz. 0.173 cbm. As the latter is represented on the theoretical diagram by the line AB 100 mm long, we only require to add to this the portion $OA = 8.5$ mm, to get OB the terminal volume of the steam. **Volume of the clearances.**

20) c. **The initial volume of the steam** is made up of the clearance plus the volume swept at the moment of cut-off. To what extent the clearance space of engines with considerable compression requires correcting in order to obtain the exact initial volume is explained in 51). **Initial volume of the steam.**

21) *The theoretical cut-off volume* follows at once from the weight of feed-water, on the assumption that the engine is working steadily, the boiler, steam-pipe, &c. are perfectly tight, and therefore that the weight of steam used and weight of feed-water are identical. By taking the feed-water from a previously measured vessel, during a period corresponding to a certain number of strokes, the weight of steam per stroke can be closely determined. For the engine we are at present considering it was 0.1425 kilos, so that its corresponding volume (as steam) was by Col. 10 of the table on p. 30 at 5 atmospheres absolute pressure, $0.1425 \times 0.3754 = 0.0535$ cbm, or in round numbers 31 % of the volume swept. So that if **Theoretical cut-off volume.**

we set off 31 mm from C to J parallel to AB, then the line CJ gives the cut-off volume and NJ the initial volume in the theoretical diagram. Other methods of computing the initial volume, particularly when combined with the determination of the degree of moisture (see 34) will be discussed later on under the different methods of investigation with the indicator.

Laws of the changes of state of the steam.

22) d. **The law of the change of state of saturated steam** during expansion has been approximately represented by means of

α) the adiabatic for saturated steam,

β) the saturation curve,

γ) the isothermal of the perfect gases.

23) *α. The adiabatic for saturated steam* has been substituted for the expansion curve of the indicator diagram, particularly by RANKINE*) and ZEUNER**). The application of the principles of thermodynamics to the process in the steam engine led both of them to the view that expansion, particularly in jacketed cylinders, must be unattended by any particular communication of heat either from or to the steam. However, experience has shown in the meantime that the expansion line of the cards from good engines is invariably higher than an adiabatic laid through the point of cut-off (see 8) and constructed according to § 6, 26 with $pv^n = const.$, for one of the usual values of n given on p. 43. The rising of the expansion curve of the indicator diagram above the adiabatic, examples of which are shewn on Plate 1, Figs. 5 and 6 also on Plate 3, Fig. 3, and Plate 4, Fig. 1, was at first attributed to the moisture of the steam, which in theory certainly makes a very close approach of the adiabatic to the saturation curve possible (see § 11, 23). But degrees of moisture up to 25 % (still assumed by ZEUNER) are not met with in practice, as related in § 12 on recent investigations of this subject. And even if they did occur, we should still have the influence of the cylinder walls upon the steam, greater in an unjacketed than a jacketed cylinder (§ 15, 24), so that strictly speaking we never have to do with adiabatic expansion. A card whose expansion line approaches the adiabatic laid through the point of cut-off and is therefore much lower than the saturation curve, points to a bad state of affairs, such as a cylinder out of truth or a leaky piston. SAMUELSON'S***) recent attempt to

*) W. J. M. RANKINE. A manual of the steam engine. London 1873. P. 383 and the following.

**) G. ZEUNER. Grundzüge der mechanischen Wärmetheorie. Leipzig 1877. P. 468.

***) A. SAMUELSON. D. w. G. der Dampfexpansion. Hamburg 1888. P. 20.

shew that the adiabatic expansion of steam in a cylinder is not only based upon science, but confirmed by practice, must also be regarded as unsuccessful, universal experience shewing the contrary. When we find that some engineers, especially English (see 55 and 58), still make use of the adiabatic as the theoretical expansion curve, it is chiefly because the theoretical diagram of energy thus produced is the smallest, so that in this way the actual indicated horse-power shews a high percentage of the theoretical power and the efficiency of the engine (as a steam-user) appears to be greater than it is in reality (compare 35, 4).

24) *β. The saturation curve* (see § 11 and 12, p. 38) would correspond exactly with the expansion line of the card if, firstly, the steam were perfectly dry, unfortunately seldom the case in marine engines, and secondly if it remained dry during expansion. But as it is partly condensed on the cylinder walls during admission, and a further portion is condensed by doing work during expansion (see § 11, 5, p. 35), a certain quantity of water must be formed which prevents the weight of steam from remaining constant. So that apart from the unmistakeable re-evaporation towards the end of the expansion, it is evident that the steam can no more expand according to this curve than according to the adiabatic. However, there are cases of very high-class single-expansion condensing engines with jacketed cylinders and very tight and rapidly-closing valve gear (such as Corliss engines) shewing cards with the expansion line approaching very closely in height to the saturation curve laid through the point of cut-off. The expansion line of the indicator cards of single-expansion condensing engines with jacketed cylinders is generally lower at its commencement than the saturation curve laid through the point of cut-off, but rises considerably above it towards the end of the stroke as shewn in Fig. 6, Pl. 2. With unjacketed cylinders this deficiency in height (compared with the saturation curve) at the beginning of the expansion and excess towards the close is much more marked, as seen in the diagrams taken at the Düsseldorf trials *). Saturation curve.

25) As the saturation curve is much used by English engineers to illustrate the losses of steam arising from internal condensation, their usual method of constructing this curve will now be described. For dry steam (whose degree of moisture is Construction of the saturation curve.

*) Die Untersuchungen von Dampfmaschinen etc. der Gewerbe-Ausstellung in Düsseldorf 1880. Aachen 1881. P. 12 and 13.

therefore O), $x = 1$ and its changes of state may be approximately expressed according to Fairbairn and Tate by the formula

$$(v - 0.41)(p + 0.35) = const. \quad \ldots \ldots \ldots \ldots \quad (90)$$

for v in cubic feet (engl.) and p in ℔s per □″ (engl.). Referred to the metrical system this formula becomes

$$(v - 0.0116)(p + 0.0246) = const. \quad \ldots \ldots \ldots \quad (90^a)$$

for v in cbm and p in kilos per □ cm. To construct the curve, the point O, (Fig. 6, Pl. 2) found as directed in 19, is to be shifted to the right 0.0116 cbm on the scale of volumes, — in the present case 6.7 mm, because 0.173 cbm = 100 mm. The point O is next to be lowered 0.0246 kilos per □ cm on the scale of pressures. As the scale here used is 15 mm per kilo per □ cm, the point is lowered $0.0246 \times 15 = 0.37$ mm below the original axis of abscissæ, as indicated by the lower dotted line in the figure. The point X, thus found, is now to be regarded as the pole from which a hyperbola is to be constructed by the method described in § 6, 16 and this will be the saturation curve. As already remarked in § 11, 23, p. 44, RANKINE and, after him, many other engineers substitute for the saturation curve a hyperbola of the form $p\,v^{\frac{17}{16}} =$ const. which pretty closely approaches the indicated expansion line for dry steam in well-jacketed cylinders.

Mariotte's curve. 26) γ. *The isothermal of the perfect gases*, or the curve corresponding to MARIOTTE'S law $p\,v =$ const. was almost universally regarded as coincident with the indicated expansion line before the development of thermodynamics. In the course of time however, repeated experiments shewed that the indicated expansion line did not evenly coincide with MARIOTTE'S curve or, in other words, rose above the saturation curve unless the steam was perfectly dry and the cylinder highly heated, whereas with dry steam in unjacketed cylinders the expansion-line only approached MARIOTTE'S curve more or less closely towards the end of the stroke. German Engineers at first attempted to explain this last circumstance by attributing it to leakiness of the slide-valve admitting fresh steam during the period of expansion, and French Engineers to the moisture contained in the steam being partially evaporated in the cylinder. But the labours of those Engineers mentioned in § 15, 1, proved that the principal cause of this phenomenon, which occurs even with perfectly tight slides and dry steam, is initial condensation and subsequent re-evaporation.

Indicator curve higher than Mariotte's. 27) The above explanations are however correct in cases where the indicator curve is higher than MARIOTTE'S curve, and either

the slide is leaky — which is evidenced by the transition from the admission to the expansion line being scarcely perceptible, or on the other hand the steam is wet, — in which case the course of the expansion line is normal for the first part of it and afterwards rises considerably above MARIOTTE'S line, as shewn in Fig. 6, Pl. 2. In any case a card, the expansion line of which is higher than MARIOTTE'S line, points to a bad state of things.

General relation of the indicator curve to Mariotte's.

28) I have found the above deductions thoroughly corroborated by the indicator cards of a large number of German war-ships. Like the diagram Fig. 6, Pl. 1 they shew that with unjacketed cylinders the expansion line, towards its termination, approaches MARIOTTE'S pretty closely, and the more so, the wetter the steam used. As the jackets are only heated by blowing steam *into* them instead of *through* them, even the line of the jacketed engines does not quite reach MARIOTTE'S line in height, but follows it much more closely than is the case with unjacketed engines as Fig. 6, Pl. 1 shews. These observations not only agree with the result of the trials of 10 engines at the Düsseldorf Exhibition in 1880*) but also of the investigations of Otto H. MUELLER**) who took more than 2000 cards from about 700 cylinders.

Application of Mariotte's law to the expansion of steam.

29) It appears then that MARIOTTE'S law corresponds best with the behaviour of dry steam expanding in a jacketed cylinder, so that it is still mostly used in practice, although not founded on any scientific basis, but merely because it happens to suit. In this work it is applied to indicator investigations and calculations of horse power &c. in accordance with the practice of German Engineers such as VOELCKERS***), SCHROETER****), and ROSENKRANZ*****), Austrian Engineers as RADINGER†), PICHLER††), and HRABAK†††) and French Engineers as LEDIEU ††††) and BIENAYMÉ†††††).

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*) Die Untersuchungen von Dampfmaschinen und Dampfkesseln u. s. w. Aachen 1881. P. 12, 6.

**) Zeitschrift des Vereins deutscher Ingenieure. 1889. P. 172.

***) J. VOELCKERS. Der Indikator. II. Edition. Berlin 1878.

****) Zeitschrift des Vereins deutscher Ingenieure. 1884. P. 191.

*****) P. H. ROSENKRANZ. Der Indikator und seine Diagramme. Berlin 1885.

†) J. F. RADINGER. Ueber Dampfmaschinen mit hoher Kolbengeschwindigkeit Vienna 1872.

††) M. Ritter von PICHLER. Der Indikator und seine Diagramme. Vienna 1880.

†††) J. HRABAK. Hilfsbuch für Dampfmaschinen-Techniker. Berlin 1883.

††††) A. LEDIEU. Les nouvelles machines marines. Paris 1876—1882.

†††††) A. BIENAYMÉ. Les machines marines. Paris 1887.

Completion of the enveloping diagram.

30) MARIOTTE'S curve being taken as the theoretical expansion curve, the calculated diagram Fig. 6, Pl. 2 is drawn by constructing the curve according to the method described in § 6, 16 and Fig. 1, Pl. 1 between the initial volume NJ and the terminal volume OB from the pole O. If the saturation curve is chosen instead of MARIOTTE'S, the initial and terminal volume remain the same, but the point X, determined according to 25, is used instead of the pole O, or if the adiabatic, — it is to be constructed from O with NJ and NO according to § 6, 26 and Fig. 2, Pl. 1. For the sake of comparison, all three curves are drawn in Fig. 6, Pl. 2. The last ordinate BL of the expansion curve gives the terminal pressure in the cylinder and completes the diagram $ONJLB$. If the ordinate AC is now drawn $ONCA$ represents the clearance.

Combination of the cards of single expansion engines.

31) **IV. The combination of the cards of single expansion engines** for the purpose of investigating their economy is effected as follows. The mean (of top and bottom) diagram is to be combined with the theoretical diagram drawn as above described, in such a manner that the atmospheric lines of both diagrams coincide and that the line of lead 01 of the indicator card also coincides with the line AC bounding the clearance in the theoretical diagram (Fig. 6, Pl. 2). As mentioned in 18) it is handiest to make both the theoretical and the actual indicator cards 100 mm long. If the latter cannot be arranged, the indicator cards must be lengthened or shortened to the 100 mm length.

Lengthening indicator cards.

32) The cards can very easily be lengthened by dividing them (Fig. 7, Pl. 2) into 10 parts with the parallel ruler belonging to the indicator and transferring the ordinates thus obtained in their proper order to the vertical lines I, II, III, &c., 1 cm apart. By joining the ends of these ordinates, the lengthened diagram shewn in the dotted line is obtained. In a similar manner indicator cards which are longer than 100 mm may be shortened. The top and bottom cards may be combined and the mean card obtained, see 13) at the same operation as the lengthening or shortening, by drawing the outline through the mean lengths of the ordinates taken off the top and bottom cards respectively.

Fulness of the diagrams.

33) When the actual indicator card is drawn in the theoretical diagram, the surface between the two lines is generally shaded, so as to shew the difference between the theoretical and actual cards, or the *loss of work* per stroke. It now remains to measure the area of the indicator card, which is most rapidly and accurately done by means of AMSLER-LAFON'S planimeter whereto

further reference will be made later on. If L_t □ mm express the area of the theoretical diagram and L_i □ mm that of the actual card (combined with the theoretical diagram as just described), then the coefficient

$$\beta = \frac{L_i}{L_t} \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (91)$$

measures the fulness of the indicator card in relation to the theoretical diagram, or the fraction of the work theoretically obtainable from the weight of steam (equal to the weight of feed-water) used, which is actually performed in the cylinder. This fraction is the "accountable" of English and American Engineers. In the combined diagram Fig. 6, Pl. 2 the fulness is

$$\beta = \frac{area\ 012345}{area\ ONJLB} = 0.55$$

i. e. of the external work corresponding to the weight of steam or feed-water used only 55 % is utilized in the cylinder, the rest is lost through cooling in the pipes, initial condensation in the cylinder, imperfect vacuum, leaks, &c. The fulness 0.55 is small and this is principally due in this engine to large clearance and low compression. If the former were here neglected, and if the clearance were not so large, β would come out about $= 0.6$, approximately corresponding to η_c (in § 17, 40) the thermal efficiency, which really expresses the same thing, as according to it in the French engines (see Table on PP. 122 and 123) about 50 to 61 % of the heat in the steam (equivalent to the weight of feed-water) was converted into indicated work. In EMERY'S experiments, which unfortunately were not carried out with the necessary scientific exactness and therefore give rather too high results, the "fulness" is, generally speaking,

for single-expansion unjacketed marine engines $\beta = 0.6$ to 0.7

„ similar engines jacketed $\beta = 0.7$ to 0.8

According to Otto H. MUELLER*) good stationary single-expansion Corliss engines give β as high as 0.85.

34) UNWIN**) lays through the point J, corresponding to the weight of feed-water Fig. 6, Pl. 2, an adiabatic JL instead of the isothermal used by most authors (see 29), draws through the axis of pressures ON a number of horizontal lines, and lays off on each of them the abscissa of the indicator card corresponding to the respective pressure il from ON as $i_1 l_1$. A Moisture of the steam.

*) Zeitschrift des Vereines deutscher Ingenieure. 1889. P. 357.

**) Engineering 1889. I. P. 522.

comparison of the area of the dotted diagram $n\, l_1\, e\, m$ thus produced shews very clearly the relation of the indicated work of the engine under investigation to that of a perfect engine giving the theoretical diagram $O\, N\, J\, L_1\, B_1$ with nonconducting cylinders, ideal valve gear, and no clearance. He then subtracts from the volume $N\, J$ of the steam equivalent in weight to the feed-water per stroke, the volume $J_1\, J$ of the jacket steam per stroke and draws thorough J_1 a saturation curve $J_1\, L_2$. By comparing $i_1\, l$ = the volume of steam shewn by the card as in the cylinder and clearance at any moment of the expansion, with the abscissa $i_1\, l_2$ (produced to meet the saturation curve $J_1\, L_2$) = the volume which would be in the cylinder if the steam remained perfectly dry during expansion, — we can infer the actual state of moisture of the steam, because the volume $l\, l_2$ not shewn by the indicator, can only be present in the cylinder in the form of water. The fraction $\frac{i_1\, l}{i\, l_2}$ is the dryness-fraction of the steam at any particular stage of the expansion (see § 11, 2 on P. 32), while $\frac{l\, l_2}{i_1\, l_2}$ gives $1 - x$, the weight of the water. Further, the ordinate $W\, l$ is the indicated pressure at any position $A\, W$ of the piston, whereas $W\, i_2$ shews the pressure which would exist behind the piston, supposing the steam in the cylinder remained equal in weight to the feed-water per stroke minus the weight of jacket-steam.

Deviations from the above combination process.

35) *Deviations from the strict method above described,* in which the indicated work was referred to the weight of feed-water are sometimes adopted, and the most usual ones are the following.

1) In place of the theoretical cut-off (see 21), the cut-off point 2 is used. Drawing MARIOTTE'S curve $D\, E$ through this latter point we obtain
$$\beta = \frac{area\ 012345}{area\ O\, N\, D\, E\, B} = 0.71,$$
i. e. the relation of the *IHP* to the theoretical performance due to the visible cut-off volume at boiler pressure. In this case the losses in the pipes (difference between the line of boiler-pressure $N\, D$ and the line of admission-pressure $n\, d$) are partially taken into account, but not those arising from initial internal condensation. The fulness β is therefore much higher than that obtained by the strict method.

2) Like 1), except that the admission pressure (see Fig. 2, Pl. 3) is used instead of the boiler pressure, the theoretical diagram is reduced by the portion $N\, D\, d\, n$, and we get

$$\beta = \frac{area\ 012345}{area\ O\,n\,d\,E\,B} = 0.75,$$

greater still.

3) In other respects similar to 1) or 2), but the saturation curve is substituted for MARIOTTE'S, as that so the theoretical diagram is smaller, and β is again increased.

4) An adiabatic is laid through point 2 and if then the virtual diminution of the clearance by means of compression is taken into account (see 51), the highest value of β is arrived at.

5) Sometimes MARIOTTE'S or one of the other curves is drawn through the point of release 3. The comparison then is between the *IHP* and the work due to the greatest weight of steam shewn by the indicator at all, as referred to in 9). In this case as before we may go back either to the boiler pressure or admission pressure.

36) From all the preceding it is sufficiently evident that when a high value of β is claimed for any engine, it does not tell us anything unless we know by which method this value has been obtained, so that the strict one which bases the calculation on the weight of feed-water is very much to be preferred to all others. Varying value of β.

37) **V. Combination of the cards of multiple-expansion engines.** Of late years the expression "*Rankinising*" for the combination of the cards of compound engines has become more and more customary in Germany, and very improperly so. The first combined diagrams were made known by COWPER, and RANKINE expressly drew attention to them six years later in a lecture treating this subject exhaustively. So that the process ought to be called "*Cowperising*" unless RANKINE'S own expression "*combination of diagrams*" is preferred. This general name is certainly better, especially as RANKINE'S particular method is at present getting more or less obsolete. Combination of the cards of multiple-expansion engines.

38) Various methods of combining the diagrams of multiple-expansion engines are in use, differing as well in the disposition of the separate cards as in the construction of the theoretical diagram enclosing them. The methods chiefly recognized are Different methods of combination.

a) COWPER and RANKINE'S,
b) WYLLIE'S, principally used in England,
c) KENNEDY'S,
d) SCHÖNHEYDER'S,
e) KIRK and PARKER'S.

Cowper and Rankine's method.

39) a. **In COWPER*) and RANKINE'S**) method,** the top or bottom cards, or preferably the mean cards of the different cylinders (see 13), are placed so that their lines of lead touch the vertical AC and their atmospheric lines coincide. The zero line OB is then drawn beneath them according to 15). Either the high or the low-pressure card may be taken first. In the former case (see Fig. 8, Pl. 2), Ac is made equal to the absolute admission pressure and the diagram $cdgh$ drawn from the point c. In the same way AP is made equal to the absolute admission pressure of the next cylinder and its diagram drawn from the point P, &c. In the present case we have the diagram of a compound engine, so that $PkQM$ is the low-pressure card.

Adaptation of the cards to the scale of volumes.

40) As a rule, the cards as taken with the indicator are of about the same length, so that in order to combine them in one diagram, they must be altered till their lengths are proportional to the volumes swept by the respective pistons. Therefore the LP diagram $PkQM$ must be extended in the ratio $\frac{V}{v} = \frac{AB}{AS}$, if $v =$ the volume of the HP cylinder (see 18) and V that of the LP cylinder. The diagram is extended as directed in 32). RANKINE divides up the diagram to be extended by a number of parallels to AB, as here drawn, and sets off on each of them the distance $rq = \frac{V}{v} rs$. Either of these methods of extension produces the LP diagram $PKEM$. For a multiple-expansion engine the diagrams which come in between the high and low-pressure diagrams are of course to be similarly adapted to the scales of volume and pressure. If an engine of this kind has two cylinders for one stage of expansion, for instance where a compound has two LP cylinders or a triple two HP ones, the volume V or v expresses the sum of the volumes of these two equal cylinders belonging to the same stage.

The scale of pressures.

41) It is evident that in the combined diagrams all the separate cards must be referred to the same scale of pressures. In the original cards this is not usually the case, as the indicator springs are taken to suit the pressures in the different cylinders (see 15). In combining the diagrams, the scale corresponding to the weakest or LP spring is generally used, for the sake

*) Transaction of the institution of naval architects. 1864. P. 248.

**) J. W. M. RANKINE. On the working of steam in compound engines. The Engineer of March 11. 1870.

of clearness, viz. 20 to 25 mm = 1 kilo per □ cm. The *LP* diagram is then drawn first and the others built up, as it were, above it.

42) Fig. 9, Pl. 2 shews the cards of the S. S. "County of York"*) built by the Barrow Shipbuilding Co., combined by COWPER and RANKINE'S method. The volumes of the cylinders are as *1* : *2* : *4* : *8* and the cards were originally taken with springs of 4, 6, 6, and 18 mm = 1 kilo per □ cm respectively. In this case it was most convenient to have the 2nd. *MP* card (v_3) in its original scales of both volume and pressure and adapt the other three cards to it. These three are divided vertically into 10 equal parts by the parallel ruler as usual. The *LP* diagram is lengthened in the proportion $\frac{v_4}{v_3} = \frac{8}{4} = \frac{2}{1}$, that is to twice its original length, and at the same time its ordinates reduced to $\frac{6}{18} = \frac{1}{3}$ of their actual height to bring them to the same scale of pressure as that of the 2nd. *MP* diagram. The 1st. *MP* diagram is shortened to $\frac{v_2}{v_3} = \frac{2}{4} = \frac{1}{2}$ its original length, whereas its ordinates remain unaltered as they are on the same scale as those of the 2nd. *MP*. Finally the *HP* diagram is shortened to $\frac{v_1}{v_3} = \frac{1}{4}$ of its length and its ordinates increased $\frac{6}{4}$, or $1\frac{1}{2}$ times. It is handier in practice to divide the cards by vertical ordinates as usual than by horizontal lines as proposed by RANKINE because the indicator scales are generally in a much simpler ratio to each other than the volumes of the cylinders. **Example.**

43) The next step to putting the cards together as above described is to construct the theoretical diagram enclosing them. The line *NJ* is drawn parallel to the zero line at the distance *AC* = the absolute boiler pressure, Fig. 8, Pl. 2. If only the admission pressure is to be taken into account (which is rarely done) this upper horizontal line is laid through the point *c* of the *HP* diagram. Then, to determine the pole *O* for the theoretical expansion curve, *AO* is made equal to the *HP* clearance on the scale of volumes. COWPER and RANKINE adopt Mariotte's curve as the theoretical curve but, as explained in 34), the adiabatic or the saturation curve is often used and sometimes both are drawn in. **Choice of the theoretical expansion-curve.**

*) Engineering 1887. I. P. 297.

Starting point of the theoretical expansion curve.

44) Strictly speaking the theoretical expansion curve should start from the point J corresponding to the weight of steam equal to the weight of feedwater per stroke (compare 35), or at any rate from the point J_1, corresponding to this weight of steam minus the jacket-steam (see 34); RANKINE however draws it through the release g of the *HP* cylinder and compares the area $ACDEB$ of the enveloping diagram thus produced, in which the area $ONCA$ corresponding to the action of the clearance is left out, with the total area of the indicator diagrams. He therefore compares the indicated work with the work which theoretically ought to be produced by the expansion of steam of the boiler pressure from a cut-off volume ND in the *HP* cylinder corresponding to the greatest weight of working steam shewn on the cards to the terminal volume OB equal to the volume swept by the *LP* piston plus the clearance of the *HP* cylinder, without taking account of the work performed by the steam in this clearance. This method is based on the assumption that the *HP* clearance steam takes part in the expansion, and does not require to be renewed from the boiler at every stroke, but is produced by the compression taking place in the *HP* cylinder. But as the work necessary to produce this compression is not deducted from the indicator diagram (see 57) the coefficient β, resulting from the comparison of the indicated and theoretical work, comes out rather too high in reality. This increase of the value of β is however so small that it can be neglected in practice.

Other starting points for the theoretical expansion curve.

45) Although the point g, the *HP* release, selected by RANKINE for fixing the position of the theoretical expansion curve is certainly the most correct if there are no means of ascertaining the actual feed-water used, many engineers draw this curve through d the cut-off point of the *HP* card, or through the point Δ Fig. 2, Pl. 3 corresponding to the initial pressure and the actual cut-off point of the valve-gear; others again, but more rarely, draw the line through the release e of the *LP* cylinder. If the theoretical expansion line is drawn through RANKINE'S point g, we may infer in general that *HP* indicator curves falling within this line point to internal condensation, those outside it to re-evaporation, and those coinciding with it to efficient jacketing. As the indicator curves of the other cylinders mostly lie inside the expansion line drawn through g, it is advisable, in order to gain an insight into the internal condensation, re-evaporation, and working of the jackets, to lay a separate MARIOTTE'S curve through the release of each cylinder.

46) In all cases where the clearances are not accurately known and we require merely a general comparison of the cards of different engines, or to know whether a set of indicator cards really belong together or not (see 60), COWPER and RANKINE'S method is preferable to all others, as the simplest and most rapid. If however we want a strict analysis of the cards, the following methods are more to be recommended. The "fulness" β of the indicator diagrams of marine engines combined by RANKINE'S method (MARIOTTE'S line laid through the *HP* release) is **Practical applicability of Cowper and Rankine's method.**

for good compound engines	0.70 to 0.75	in rare cases.
" " triples	0.70 " 0.85	
" " quadruples, so far as known hitherto	0.60 " 0.66.	

47) b. **The method which WYLLIE*)** and many other English engineers employ only differs from RANKINE'S in this respect that each of the different cards (Fig. 2, Pl. 3) is placed with its admission point at the distance corresponding to its clearance volume OA or OA_1 from the axis of ordinates ON. The theoretical diagram is described as before about the combined indicator cards, which in the present figure are those of the compound engines of H. M. Cruizer "Boadicea" by RENNIE**) with one high and two low-pressure cylinders. Either of the three theoretical expansion lines referred to in 22) may be drawn through either of the points before described. In the figure all three are shewn and the point $\varDelta$ is used, corresponding to the initial pressure and the actual cut-off point of the valve-gear. The most usual is MARIOTTE'S line laid through the *HP* release and carried up to the boiler-pressure. The area of the theoretical diagram then represents the external work produced by a volume of steam equal to a certain cut-off volume of the *HP* cylinder plus the clearance of this cylinder, expanding from boiler pressure to the volume swept by the *LP* piston plus the clearance of the *LP* cylinder. As before, the "fulness" β is the quotient of the combined areas of the indicator cards divided by the area of the theoretical diagram. But as this area does not differ essentially from that produced by RANKINE'S method, this method is handier on account of its simplicity, because the clearances are seldom known by measurement. When this is the case, however, WYLLIE'S method is to be preferred, as the more exact one. **Wyllie's method.**

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*) Proceedings of the institution of mechanical engineers. 1886. P. 473.

**) Transactions of the institution of naval architects. 1874. P. 156.

Kennedy's method.

48) c. **The method which KENNEDY*)** first used for compound engines in comparing steam and gas engines with the object of deciding upon the most economical motor for dynamos, and afterwards on his well-known trials of the S. S. "Meteor"**) (see § 17, 38, p. 142) is almost identical with the last-described method as is at once seen from Fig. 1, Pl. 3, the diagram of a two-cylinder compound non-condensing engine. But KENNEDY bases his theoretical diagram upon the weight of feed-water used per stroke, so that NJ represents the volume of steam of this weight at boiler pressure. Through J a saturation curve JL is drawn completing the enclosing diagram $ONJLB$, the area of which divided into the combined areas of the indicator diagrams gives the "fulness" β. The dotted diagrams in the figure shew by their abscissæ for any particular pressures the ratio of the volume of the visible steam to that corresponding to the weight of feed-water. By what is said in 34) however, it is evident that KENNEDY'S method is not quite correct. His choice of the saturation curve for a theoretical expansion curve instead of the more usual MARIOTTE'S curve merely increases the value of β, as shewn in 35), 3. But his disposition of the dotted diagrams so as to touch ON, in order to determine from them the curve of the visible weight of steam is, by 34), so far inaccurate as it leaves out of account the whole of the clearance steam. This arrangement of the diagrams may facilitate the comparison of the indicated with the theoretical work, but cannot be adopted if we wish to ascertain the moisture or in other words the dryness fraction of the steam, as this can only be seen by the difference between the ordinates at any particular volume of the saturation curve JL and the expansion curve of the indicator diagram. As before mentioned, NJ must represent the weight of steam equal to the weight of the total feed-water per stroke in unjacketed engines, and for jacketed ones the above weight minus the weight of jacket steam per stroke. In spite of these incorrectnesses, KENNEDY'S method is more exact than the two first described. If we are once to take the trouble to measure the clearances we ought not to be deterred from ascertaining, whenever it is possible, the feed-water and jacket steam per stroke, so that by means of the former we may compute from MARIOTTE'S curve laid through J, Fig. 6, Pl. 2 the only correct value of the "fulness" β, and by means of the

*) Journal of the Society of Arts. February 15. 1889.

**) Proceedings of the institution of mechanical engineers. London. May 1889.

latter (the distance JJ_1 in the same figure) we may draw the saturation curve $J_1 L_2$ shewing the dryness-fraction at any point.

49) d. **SCHÖNHEYDER'S method, also used by KRAUSE,*) MUDD,**) and CAMPBELL***)** is distinguished from the foregoing by each of the diagrams being placed at a distance from the axis of ordinates ON, equal, not to the clearance of its own cylinder but to the compression volume of the preceding cylinder, and by each card having a separate theoretical diagram to itself instead of their all being referred to one common one. The philosophy of this method may be explained as follows. **Krause and Mudd's method.**

50) A non-condensing engine without clearance or compression gives an indicator diagram $GCDEF$ Fig. 10, Pl. 2, the expansion curve DE of which is a MARIOTTE'S line drawn through D from the pole A and corresponding to the cut-off CD. But if the cylinder has a clearance of the volume AO, the initial volume is ND and the expansion curve must be drawn from the pole O and takes the course DH. So that the cylinder with clearance does the work DEH more than the cylinder without clearance. If the exhaust is closed at J, and the steam remaining in the cylinder compressed up to the boiler pressure, the clearance is already filled with steam at the beginning of the next stroke and only the volume CD requires to be supplied from the boiler, whereas the whole volume ND expands according to the curve DH. The original indicator diagram $GCDEF$ is then changed into the diagram $JCDHF$, i. e., it is augmented by the area DEH corresponding to the increased work of expansion and diminished by the area GCJ which expresses the work of compression. If DEH were always equal to GCJ which is the case when the pressure BH at release equals the back-pressure BF, the diagram $GCDEF$ could be substituted for the diagram $JCDHF$. **Influence of the clearance and compression upon the expansion.**

51) In condensing engines compression cannot be carried far enough to bring the pressure in the cylinder equal to the boiler pressure. If for instance the compression begins at K and rises to the pressure AP, the clearance is filled at the beginning of the next stroke with steam of low pressure, and a certain quantity of boiler steam must pass into it to increase the pressure AP to the initial pressure AC. To deter- **Exact determination of the steam per stroke.**

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*) Zeitschrift des Vereines deutscher Ingenieure. 1886. P. 596.

**) The marine engineer. 1887–88. P. 58.

***) Ibid. 1888–89. P. 155.

mine this volume of steam which must be admitted to the clearance, produce the compression line KP to S, then CS expresses the volume required. In accurate investigations the compression line KS should be drawn as an adiabatic from the pole O (see 12). If however the exact form of the compression line is not of importance, we may assume it to be a MARIOTTE'S curve and the distance CS may be determined by dropping the perpendicular Kk_1 from K and drawing a straight line $k_1 O$ from O, the intersection Q of which with the back-pressure line GF of the card, gives the volume sought $SC = QG$. The area $QSPG$ now expresses the work which the steam entering the clearance must perform in order to equalize the pressure in it, and $SC + CD$ is the total volume of steam required to be supplied from the boiler per stroke. This expands (without regard to the volume NS which always fills the clearance) down to the terminal volume $O_1 B$ equal to the volume AB swept by the piston plus that portion AO_1 of the clearance which is filled with *fresh* steam at each stroke. So that O_1 now becomes the pole for the construction of the expansion curve DM and we get the diagram $KSDMF$ which takes the compression-work SQK into account. See further on this subject in 57).

Succession of the diagrams.

52) SCHÖNHEYDER describes a diagram of this kind about the card of *each* cylinder (Fig. 1, Pl. 4 — MUDD'S combined diagram of the quadruple-expansion engines of the S. S. "Suez"*). The absolute boiler pressure $A_1 C_1$ is set off upon the axis of ordinates, the upper horizontal line $N_1 D_1$ drawn, and upon this the *HP* clearance $C_1 N_1 = A_1 O'$ set off to the left. The *HP* diagram is now placed with its line of lead touching the vertical $A_1 C_1$ and the horizontal line $S_2 D_2$ drawn through S_2, the beginning of the compression. It now remains to draw the vertical line $S_2 k_1$ through S_2, also to draw the line $k_1 O'$, cutting $S_2 D_2$ in Q_1 and the perpendicular $S_1 O_1$ dropped from this point gives the pole O_1 for constructing the theoretical expansion curve $D_1 D_2$, while O' remains the pole of the theoretical compression curve $S_1 S_2$. MUDD adopts MARIOTTE'S curve for both, and drawing the former through the cut-off point of the *HP* diagram, gets the theoretical cut-off volume $C_1 D_1$. Of course any other theoretical expansion line may be used and drawn through any other of the points referred to in 45). As shewn in 51) the adiabatic is probably the best for the theoretical compression curve.

*) The Engineer. 1888. I. P. 162.

53) The point S_2 fixes $S_2 D_2$, the volume of steam which passes from the first cylinder to the second and $S_2 X_1$ the volume left in the first. $S_2 D_2$ is therefore the actual initial volume of the second cylinder, i. e. the volume $S_2 C_2$ necessary to equalize the pressure in its clearance plus $C_2 D_2$ which fills the cylinder itself up to the cut-off point. In order to determine the distance $C_2 S_2$ at which the diagram of the second cylinder is to be placed from the perpendicular $S_2 k_1$ (which depends upon the compression in the first cylinder), we must, before drawing the diagram of the second cylinder, set off the volume of its clearance $C_2 N_2 = A_2 O$ from $C_2 A_2$, draw the perpendicular $S_3 k_3$ and the straight line $k_3 O''$, the intersection Q_2 of which with the horizontal line $S_3 D_3$ drawn through the compression point S_3, then gives the distance $Q_2 X_2 = S_2 N_2$ and therefore also $S_2 C_2 = C_2 N_2 - S_2 N_2$. The perpendicular $Q_2 O_2$ drawn through Q_2 fixes O_2 the pole for the theoretical expansion line and O'' becomes the pole for the theoretical compression line $S_2 S_3$. Now the second diagram can be drawn in its proper position with regard to the first and those of the other cylinders follow in a similar way. The cross-hatched surface in the figure shews the theoretical diagrams bounded only by the expansion and compression curves of the different cylinders, and corresponds to the work performed in them by the steam admitted per stroke, if the steam always remaining in all the clearances is neglected and the initial volume is assumed to be the *visible* cut-off volume $S_1 D_1$ of the *HP* cylinder.

Distance of the different diagrams from the axis of ordinates.

54) As the four different poles $O_1 O_2 O_3 O_4$ are used for the expansion lines of the four cards, and the poles $O' O'' O''' O''''$ for the compression lines, these curves cannot run fairly into one another, but shew as many breaks as there are cards. The diagram illustrates clearly the divergence of the different expansion lines from the ordinary MARIOTTE'S curve $D_1 H$ drawn through D_1 from the pole O'. The compression line $C_1 J$ is also drawn in, representing that of an equivalent single-expansion engine in which the compression is carried up to boiler pressure. The area contained between $C_1 J$ and the separate theoretical lines $S_1 S_2 S_3 S_4 S_5$ shews *that in a multiple expansion engine the action of the clearances is such that less compression work is lost than in an equivalent single-expansion engine.* It is similarly shewn by the portion of the theoretical *LP* diagram to the right of the perpendicular HB, the length of which corresponds to the distance $A_1 A_4$, *that in the multiple-expansion engine, in consequence of the extended expansion, more*

Divergence of the theoretical curves of the different cards in Schönheyder's method.

expansion work is gained than in the equivalent single-expansion engine. It is true that from this gain of expansion work the narrow and unimportant area lying between $D_1 H$ and the broken expansion line $D_1 D_2 D_3 D_4 D_5$, must be deducted as being in fact an excess of expansion work in favour of the single-expansion engine, but a comparison of the two areas shews that the advantage of the multiple-expansion engine is not materially reduced by it. In a compound non-condensing engine with 7 kilos per □cm initial pressure (say 105 ℔s), expanding five times, the advantage from reduced compression and increased expansion is theoretically about 5 % compared with an equivalent single-expansion engine, the compression being in each case carried up to the initial pressure.

Inferences from the combined diagram.

55) Those portions of the different indicator cards which extend outside the theoretical expansion-curves $D_1 D_2 D_3 D_4 D_5$ point to very considerable re-evaporation in the respective cylinders and this is particularly the case in the midde ones, which MUDD claims to have achieved in the present engine (S. S. "Suez") by jacketing and highly heating the receiver walls. He calculates the percentage of efficiency or "fulness" of the combined diagrams for the theoretical diagram $A_1 C_1 D_1 E B$, which like RANKINE'S neglects the clearance and has an adiabatic $D_1 E$ laid through the *HP* cut-off as its expansion-line, at 76.9 %. If however we take MARIOTTE'S curve $M_1 M$ instead of the adiabatic and draw it through the *HP* release as usual, this percentage shrinks to 6.

Applicability of Schönheyder's method.

56) This, the exactest of all methods, should by rights always be used in combining the diagrams of compound and multiple-expansion engines, particularly when there is considerable compression in any of the cylinders. The reason why this method is nevertheless rarely adopted in practice is chiefly because the volumes of the clearances are seldom accurately known. Besides, the process is complicated and absorbs too much time compared with COWPER and RANKINE'S very simple one. Considering also that MARIOTTE'S curve which is used for the theoretical expansion and compression lines is, after all, not quite correct, and that with moderate compression and fifteen or more expansions, which are usual in triples and quadruples, the gain from less compression and greater expansion compared with an equivalent single-expansion engine is only very slight, — it cannot be denied that COWPER and RANKINE'S method is nearly as accurate as SCHÖNHEYDER'S. For judging of the action of the steam in the separate cylinders SCHÖNHEYDER'S method is certainly preferable to the

others hitherto described. But for investigating the influences of moisture of the steam, condensation during expansion, and the jacketing of the cylinder and receiver walls, this method can only be applied after setting off from the compression points $S_2\ S_3\ S_4\ S_5$ the volumes $S_2 F_2$, $S_3 F_3$, $S_4 F_4$, &c. representing the volumes of steam (at the pressures of the respective points) of equal weight to the feedwater per stroke less the jacket steam per stroke, and drawing the corresponding saturation-curves to the points $F_2\ F_3\ F_4\ F_5$.

57) e. **KIRK*) and PARKER'S method** is illustrated in Fig. 10, Pl. 2. As mentioned in 51) $KSDMF$ is a diagram which takes into account the compression work as well as the expansion work of the fresh steam entering the cylinder at each stroke. The area PSC, extending outside the stroke AC of the indicator diagram, represents the work which would have to be done in the cylinder to bring up the pressure in the clearance to the initial pressure, without any fresh steam entering. So that in order to use DM as the theoretical expansion line, the outside surface $PSC = PRC$ must be deducted from the indicator diagram, leaving $KPRDMF$. **Theoretical basis of Kirks method.**

58) Starting from this, KIRK and PARKER place their indicator cards, like RANKINE, against the perpendicular AC (Fig. 5, Pl. 3, combined diagram of the triple-expansion engines of the S. S. "Aberdeen"), set off the clearance-volumes to the left, construct in these (by 51) the compression-lines KPS (see diagram of the intermediate cylinder), and deduct the area $PSC_1 = P_1 C' R$ from the indicator-diagram. As a theoretical expansion-line KIRK draws an adiabatic DE of the form $pv^{1.9}$ $= const.$, through the cut-off point 2 of the *HP* cylinder, whereas PARKER draws the same curve through the *HP* release 3. In the figure a MARIOTTE'S line $3E_1$ is drawn through this latter point to illustrate the effect upon the "fulness" (or percentage of efficiency) of drawing an adiabatic through 2 or an isothermal through 3. OA is the *HP* clearance-volume and O is the pole of the adiabatic. The figure shews that the correction for compression work in the clearance is unimportant in the *HP* and *MP* cylinders and only affects the *LP* cylinder in any serious degree. KIRK'S method is thus rather more accurate than RANKINE'S certainly, but is decidely inferior to SCHÖNHEYDER'S, especially when the latter is used in combination with saturation-curves as is done by UNWIN. **Construction of the combined diagram.**

*) Transactions of the institution of naval architects. 1882. Plates I and III.

Further information obtained from combined diagrams.

59) **VI. Further points of interest connected with combined diagrams.** Besides elucidating the efficiency of an engine as a steam-user by the percentage β of the area, the average values of which are given in 33) and 46), illustrating the effects of condensation, re-evaporation, and jacketing, upon the configuration of the indicator diagrams as compared with saturation curves, the combination of the cards of multiple-expansion engines affords us further the means of ascertaining

a) whether any given set of cards belong to each other or not,
b) the influence of admitting or withdrawing any steam to or from any of the receivers between the cylinders,
c) the influence of the velocity of the steam.

Investigation of the connection between the cards of a set.

60) a. *The connection between the cards* must first be established before the alleged indicated horse-power of a new engine can be accepted or any calculations as to its economy be gone into. In exact trials of multiple-expansion engines the cards beginning at the *HP* cylinder ought properly to be so taken they actually represent the work of one identical volume of steam on its way through the successive cylinders. As this is unattainable with fast-running engines, the cards are taken on all the cylinders simultaneously, on the assumption that in a "permanent state" the phenomena in all the cylinders do not vary during a very short space of time. A piece of bad practice in this respect has however, unfortunately crept in, some firms having, in order to shew a high *IHP* and economy of fuel, picked out of a number of cards taken at different times, the largest they could find of all the cylinders and afterwards combined them. Whenever combined diagrams prepared in this way are published, it can easily be shewn by either of the methods just described that the several cards do not belong to each other. If the admission line of any card begins higher than it could if it corresponded to the back-pressure line of the preceding card after allowing for the loss of pressure between the cylinders, as is the case in the combined diagram of the "Rionnag-na-Mara"*), Fig. 4, Pl. 3, in both of the intermediate cylinders, it is a sure sign that the cards do not belong to each other and that the intermediate diagram is too large. The *IHP* calculated from these cards which do not belong to any one simultaneously-taken set is therefore greater than the average power during the trial and of course the consumption of coal, referred to this power, is too small.

*) Engineering. 1886. I. P. 361.

61) b. *The influence of the admission or withdrawal of* steam between the cylinders is shewn, in the former case, by the *MP* and *LP* diagrams being, when combined obviously much too large for the *HP* one. An instance of this is given in Fig. 3, Pl. 3, a diagram of the engines of the S. S. "Arabian" *) shewing that the steam in the *MP* steam chest was "freshened" with direct boiler steam, which however is not mentioned in the report. The horse-power calculated from these cards is too high and the consumption of coal, probably measured at a time when the direct connection with the *MP* steam chest was shut off, too low. — The second case, that of the withdrawal of steam, is shewn by the S. S. "Aberdeen" **) Fig. 3, Pl. 3, steam being here taken from the receiver between the *MP* and *LP* cylinders for a WEIR'S heater. On this account, the *LP* diagram exhibits a considerable fall below the adiabatic laid through the cut-off point of the *HP* cylinder, which is still more clearly brought out on taking MARIOTTE'S line drawn through the *HP* release as the theoretical curve. The loss of work due to the steam employed in the heater is partly recovered in the increased temperature of the feed.

Influence of admission or withdrawal of steam between the cylinders.

62) c. *The influence of the velocity of the steam* or, in other words, of any obstruction in the pipes and passages may be judged of by the deviation of the admission and back-pressure lines from the horizontal, a considerable deviation evidencing decidedly unfavourable conditions. A series of such cases, compiled by WYLLIE ***) from his own experience is here briefly referred to as being very useful for comparison.

Influence of the velocity of the steam.

1) The diagram Fig. 2 Pl. 4 is taken from the *HP* cylinder of a triple-expansion engine working at 143 ℔s, to the main steam pipe of which several stop-valves were fitted for various subsidiary purposes, in consequence of which the steam lost 15 ℔s pressure by the time it reached the cylinder. The diagram next to this, Fig. 3, shewing 138 ℔s initial pressure, was taken after the valves were removed from the main steam pipe. The piston speed of this engine was about 600 ft. In another triple-expansion engine of the same dimensions there was a fall of only 5 ℔s from the boiler to the steam chest at 153 ℔s boiler pressure and 420 ft piston-speed (Fig. 4), whereas with increased piston speed the diagram Fig. 5 was produced, shewing considerable wire-drawing.

*) Engineering. 1884. II. P. 84.

**) Transactions of the institution of naval architects. 1882. P. 36.

***) Proceedings of the institution of mechanical engineers. 1886. P. 476.

2) The *MP* diagram Fig. 6 is from a triple fitted with piston-valves which throttled the steam in the indirect passages, while Fig 7 is from an exactly similar engine with ordinary slides and rather lower piston speed, shewing a great reduction in the wire-drawing.

3) The influence of the velocity of the steam is illustrated by the *MP* diagrams of the same engines, Figs. 8 and 9, Pl. 3. The former was taken on a light trial trip with the engines running fast, the latter when the ship was loaded and the engines making fewer revolutions. The cut-off was the same in both cases, but the *MP* receiver pressure under the latter conditions is about 7 ℔s less and the indicator curve approaches the theoretical curve more closely.

4) The desirability of keeping the *LP* clearance as small as possible has a tendency to lead to the ports being made too narrow for high piston speeds. Fig. 7, Pl. 3 is a diagram from such a cylinder taken at 78 revolutions, the engines having been originally designed for only 60 to 65 revolutions. The vacuum at 68 revolutions, as Fig. 6 shews, is nearly 2 ℔s better, the resistance of the eduction pipe being less on account of the reduced velocity of efflux of the steam.

Value of indicator trials.

63) From the preceding it is evident how valuable indicator trials and the investigations based upon them are, particularly with multiple-expansion engines, on account of the manysidedness of the practical information they afford. It therefore appears justifiable to devote so large a space to this subject.

Fourth Division.

Fuels.

§ 19.

Fuels and their qualities.

1) For marine boilers there have been used or tried Divisions of the subject.

I. Solid fuels,

a) Wood,

b) Turf and peat,

c) Lignite or brown coal,

d) Coals and Anthracite;

II. Liquid fuels,

e) Petroleum and its products of distillation.

f) Tar and tar-oil,

g) Shale-oil;

III. Gaseous fuels,

h) Gases from solid fuels,

i) Gases from liquid fuels.

2) **I. Solid fuels** are for the most part burnt in marine boilers, Solid fuels. the quantity of liquid fuels used is comparatively insignificant, and gaseous fuels have only been experimentally applied. All solid fuels contain, besides their combustible components — principally carbon, hydrogen, and oxygen — certain non-combustible parts — *ash*, — and *moisture or water*, taken up from the atmosphere &c., the proportion of which affects the calorific value of the fuel. The table on page 182 shews the compositions and values of the fuels now to be described.

3) a. **Wood** forms the fuel of some steamers in countries rich in Wood. timber and poor in coals, for instance on several rivers in Eastern Russia, Siberia, and Central Africa. The mean percentage of water in perfectly seasoned pine is about 15 and that of other woods about 17 according to CHEVANDIER'S researches; it is generally taken at 20 % average, and in newly-

felled timber at 40. All pines have the smallest percentage of ash — between 0.3 and 0.6 %, in other woods this varies from 0.8 to 1.2 % and is generally assumed in practice to be 1 %. BRIX'S *) experiments shewed that the calorific value of different woods may be taken as approximately proportional to their specific gravity when their percentage of moisture is the same, and only very mature pines form a favourable exception to this rule. About 2.5 to 4, or on an average 3 kilos of water at 0^{0} can be converted into steam at 100^{0} with 1 kilo of seasoned wood. Of all woods, pines are most easily ignited and burn with the longest flame, on which account they are preferred for lighting up the fires. In this respect birch comes next to pine and may be used as a substitute for it. One metrical ton = 1000 kilos of cleft birch or beech takes up about 2.5 to 3 cbm in the bunkers; if very knotty and crooked, rather more.

Turf.

4) b. **Turf and peat** are only used in very rare cases as an auxiliary fuel in the boilers of a few river steamers. They are a product of the spontaneous decomposition of plants, especially those inhabiting marshy regious. The younger and fibrous layer is called *turf*, the older, heavier, and drier stratum be neath it, *peat*. Air-dried turf retains in general about 25 % of water and its percentage of ash varies from 1 to 30 %, the latter may be taken on an average for turf, to be profitably used as fuel in boilers, at 5 to 8 %. Turf of medium weight takes up about 2.5 cbm per ton of 1000 kilos in the bunkers when well trimmed.

Lignite or brown coal.

5) c. **Lignite or brown coal** is used in river steamers in Central Germany and Bohemia. Like turf, it is wood or woody fibre changed by wet decay, but its process of decomposition is much further advanced than that of turf. According to its age it is distinguished as *fibrous*, *earthy*, and *shelly lignite*, of which the first-named is the youngest formation. Only the oldest of the three can be used as fuel for boilers. Air-dried lignite contains 20 % of water, but when freshly raised this percentage is as high as 30 on an average, sometimes even 50. Nevertheless freshly raised lignite is preferable to that which has been kept, as the latter undergoes a slow decomposition under the influence of air and moisture which detracts from its calorific value. Lignite contains 5 to 10 % of ash. With 50 % of moisture 1 kilo of lignite will convert 2 kilos of water at 0^{0} into steam at 100^{0}, and with less than 25 % of moisture,

*) P. W. BRIX. Untersuchungen über die wichtigeren Brennstoffe des preussischen Staates. Berlin 1853. P. 40.

4 kilos. According to its density it takes up from 1.5 to 2 cbm per metrical ton of 1000 kilos in the bunkers.

6) d. **Coal** is almost the only fuel used for seagoing steamers. It is a combustible fossil formed by very slow decomposition of woody fibre under heavy pressure and exclusion from air. It has been deprived of its oxygen in the course of ages by the formation of water and carbonic acid, while its proportion of carbon has gradually increased. The final product of the slow decomposition of which lignite and coal denote successive stages, is called *anthracite*. **Coals.**

7) From a technical point of view, coals are classified according to **Classification.**
   α) their behaviour on the grate,
   β) their percentage of oxygen,
   γ) their percentage of bitumen.

8) α. *With regard to their behaviour on the grate* they may be divided as **Behaviour on the grate.**
   anthracite, which in burning splits or flies into small pieces not adhering together;
   dry bituminous, which adheres together in larger pieces and does not split, while the smaller pieces agglomerate without however swelling or "caking";
   bituminous caking coal, which swells and melts into a paste or "cake".

9) β. *In relation to the oxygen they contain, which is also a measure of the age of their formation* and is generally a test of their other qualities, HILT*) classifies coals as follows, **Proportion of oxygen.**

17 %	of oxygen and above —	gassy dry coal,
14 to 17 %	" " "	" bituminous,
10 to 14 %	" " "	" gassy caking,
7 to 10 %	" " "	" caking,
3 to 7 %	" " "	" caking, semi-bituminous and anthracite.

For gassy coals this classification is pretty close, but becomes uncertain for caking coals and ultimately ceases to apply. It is however very important from an engineering point of view to distinguish between caking coal, semi-bituminous and anthracite. Coals which are richer in oxygen and burn with a long flame are mostly designated in practice as "*flaming*".

10) γ. *The classification according to the proportion of bitumen*, i. e. the volatile products forming flame, or more correctly, according to the ratio of the weight of these products to the weight **Proportion of bitumen.**

*) R. v. WAGNER. Handbuch der chemischen Technologie. Bearbeitet von Ferd. Fischer. 12th Edition. Leipzig 1886. P. 1011.

of the residual coke is also not entirely certain, but is a better guide in judging of the value of coals. According to Hilt we have for 100 parts of coke exclusive of ash

44.4	to	48.0 %	of	bitumen	from	gassy sand-coals,
40.0	„	44.4 „	„	„	„	gassy (young) semi-bituminous,
33.3	„	40.0 „	„	„	„	caking gas coal,
15.5	„	33.3 „	„	„	„	caking coal,
10.0	„	15.5 „	„	„	„	non gaseous (older) semi-bituminous,
5.0	„	10.0 „	„	„	„	hard anthracitic coal.

But this system of division does not classify with sufficient exactness the caking coals which may differ as much as 18 % in their gaseous products, whereas the coals of each of the other classes only very 4 to 5 % among themselves.

Designations.

11) In practice it is usual to call coals containing

over	30 %	of	bitumen	bituminous,
from	30 to 15 %	„	„	semi-bituminous,
„	15 to 10 %	„	„	slightly bituminous,
under	10 %	„	„	non-bituminous.

Further it is usual to designate as

Long flaming coals	Sandy — the dry bituminous, Semi-bituminous — the clinker forming bituminous, Gas coals — the caking bituminous.
Short flaming coals	"Fettkohlen" — the caking, semi-bituminous, "Esskohlen" slightly bituminous, Anthracites, hard non-bituminous.

Smith's coals are caking coals about half way between gassy and greasy coals.

Distribution of coals.

12) The upper Silesian coals are mostly sandy, but gas coals and semi-bituminous coals are got at Waldenburg. The Saar coals are also generally sandy and semi-bituminous as well as slightly caking and gassy. The upper Westphalian basin produces gas-coals. Besides these long-flaming coals, short-flaming ones do not occur in Germany except in the lower and middle portions of the Westphalian basin and in the Wurm and Inde region at Kohlscheid and Eschweiler. Scotch *Bog Head* coal is very bituminous and allied to *cannel* which latter is again closely related to shelly lignite. Scotch *splint* is mostly sandy, *Newcastle coals* are caking and *Welsh coals* resemble "Esskohlen".

Average composition of coal.

13) *The chemical composition of coal* varies very considerably according to the age of its formation, as shewn on the table on p. 182. The average composition of superior steam coal is

Carbon	80 %
Hydrogen	4 "
Oxygen	8 "
Nitrogen	1 "
Sulphur	2 "
Water	3 "
Ash	2 "

Coals stored in the open air generally contain about 5 % of water; the ash varies from 2 to 7 %. One kilo of coal converts 7 to 9 kilos of water into steam at 100° according to its proportion of carbon. One metrical ton (= 1000 kilos) takes up from 1.2 to 1.3 cbm or on an average 1.25 cbm in the bunkers. In England 42 to 45 cubic feet of bunker space are allowed for 1 ton (2240 ℔s). A cubicmeter of "teamed" coal weighs from 700 to 900 kilos but with very light kinds only 650, while very heavy coals will weigh as much as 920 kilos per cbm.

14) *Compressed coal or "patent fuel"* consisting of small coal and some binding medium and formed into cakes or "bricquettes" in a press is largely used in modern war-ships as it is so well adapted for stowing in the cellular divisions above the armoured deck. The binding material for the small coal is either organic as coal tar, pitch, asphalte, starch, albumen, molasses &c., or inorganic as clay, plaster of Paris, alum and chalk, silicate of potash, &c. The chief obstacle to the use of organic materials is their high price, and the objection to the inorganic ones is their low binding power, on account of which they require to be used in such large quantities as to considerably raise the percentage of ash and thus reduce the calorific value of the fuel. In order to protect the bricquettes from damp they are sometimes immersed in silicate of potash or a solution of resin in benzine. The Westphalian patent fuel has occasionally given very good results, some kinds having evaporated rather over 9 kilos of water per kilo from 0° at 100°. One cbm of patent fuel stowed in tiers weighs between 1000 and 1100 kilos. The compressive strength of patent fuel exceeds that of the superior kinds of Westphalian and English coals used at sea. **Compressed coal or "patent fuel".**

15) **II. Liquid fuels** are adopted on the steamers of the Caspian and the lower Volga where the petroleum from Baku is cheap and the price of coals is high. A few steamers on the Black Sea are also fired with liquid fuels. In North America too they are used here and there, their extra cost being made up by the **Where used.**

reduced number of hands required in the stokehole as compared with coals. In England and France repeated attempts have been made to introduce liquid fuel for steamers, but have hitherto always failed in competing with coals from the one cause — its high price. It now appears however to have some prospect in these countries also of being used, but only for torpedo boats.

Kinds.

16) e. **Of earth-oil and its products of distillation** the following have been used as fuels,

α) Crude earth-oil or kerosene,

β) The residuum of the distillation of lamp oil from kerosene,

γ) „ „ „ „ „ „ lubricating „ „ .

Qualities.

17) α. *Crude kerosene* was formerly frequently burnt in locomotive and marine boilers in Pennsylvania and has been again used in some recent American experiments. One of the chief objections to the American crude oil is that it contains as much as 20 % of its volume of limpid and easily-inflammable oils, which bring down its "flashing-point" to 15° and therefore render great care necessary in storing it on board. But the principal reason for not adopting this oil as fuel is its high price, which even at the bore-hole is about twice as high as that of coals. The Caucasian oil on the other hand, is at its source much cheaper than coals and was much used on the Caspian steamers until the Russian Government forbade its further application to this purpose on account of repeated fires on board ship due to its inflammability. As the crude Caucasian oil only contains on an average 5 to 6 % of thin oils which can be almost entirely volatilized by storing it for a few days in open vessels, thus reducing its weight 10 to 15 %, its flashing point is above 20° and it is therefore far less dangerouss than the Pennsylvanian oil. The veto of the Russian Government has hence often been denounced as a sin against industry. — Crude earth-oil from Rangoon which is heavy and viscous was experimented on as a fuel at Woolwich in 1864 by RICHARDSON, but it had to be mixed with lighter oils to render it more limpid and suitable for the purpose. Apart from its much disputed danger, crude petroleum is not a particularly economical fuel, experience shewing that the more limpid the oil is, the greater is the consumption and the longer the flame, so that the latter under some circumstances extends above the top of the funnel. — The proportion of water in crude oil and in its residual products of distillation is estimated at about 2 %. The average measurement of a

metrical ton (1000 kilos) of all the different oils may be taken at 1.1 cbm stowage, crude tar however requires only from 0.85 to 0.9 cbm. All liquid fuels except crude tar (see 20) may be taken as capable of evaporating 12 to 16, or on an average 14 kilos of water per kilo from 0^0 at 100^0, whereas coals will only evaporate from 7 to 9 times their weight of water.

18) *β. The residual products of the distillation of lamp oil from crude petroleum* form the almost exclusive fuel for steamers on the Caspian and lower Volga aince the use of the crude oil was prohibited. These residual products, called in Russia "*Astatki*" amount in the case of Caucasian oil to 50 to 60 % of the crude oil worked, whereas they are only 5 to 10 % of the American oil. According to the extent to which the distillation is carried, the flashing point of the residuum is a little under or over 100^0, thus its stowage on board requires comparatively little care. The price at Baku of the "Astatki" was in 1886, according to ENGLER*) 4 s. to 5 s. per 1000 kg, so that it is an extraordinarily cheap fuel. **Quantity and price of lamp oil residuum.**

19) *γ. The residuum of the distillation of lubricating oil* will in future probably replace the residuum of lamp-oil, as it is becoming more and more usual to submit this latter residuum to a second process of distillation, in order to obtain lubricating oil from it. At present the lubricating oil residuum amounting to 20 or 30 % of the crude oil is allowed at Baku to flow into the sea unused. It has however been already applied as fuel in several places. Its viscous nature, almost solid in cold weather, necessitates either an admixture of refined petroleum, or heating by means of a steam coil, in order to obtain the requisite fluidity. As it is an almost valueless waste-product and therefore a very cheap fuel, this residuum will come more and more into use the dearer the lamp oil residuum becomes. **Prospects for lubricating oil residuum.**

20) f. **Tar and tar-oil** have a prospect of being burnt to a small extent in marine boilers at the present very low price for crude coal tar of 15 s. to 20 s. per ton at the gas-works. In England, SADLER & CO.**) have already fitted an arrangement for using tar fuel. At the annual meeting of German gas and water engineers at Eisenach in 1886 KOERTING***) described the furnaces which have been introduced in English and German gas-works for using their own bye-product, tar, as a fuel **Experiments with tar.**

*) C. ENGLER. Das Erdöl von Baku. Stuttgart 1886. P. 50.

**) Transactions of the north-east coast institution of engineers and shipbuilders. 1886. II. P. 32.

***) Journal für Gasbeleuchtung und Wasserversorgung. 1886. P. 543.

to heat the retorts. As one kilo of tar containing 8 % of water developes 8660 thermal units, it possesses a calorific value exceeding that of the best Westphalian and Welsh coals, and it is therefore not impossible, if the price of tar should fall still lower, that a few steamers in some of the great ports may be more cheaply and advantageously fired with crude coal-tar than with coals. But this made of firing can never become important on account of the small quantity of tar available for the purposes.

Experiments with tar-oil.

21) *Tar-oil* and particularly *creosote-oil*, the heavy product of a second distillation of tar after the light oil *benzol*, so much used in the manufacture of anilin dyes, has been driven off, have been the subject of many experiments by AUDOUIN *), a gas-engineer in France and Admiral SELWYN **) in England, who have sought to introduce these oils as fuel. But on account of its many uses in the arts the heavy oil is worth 40 s. per ton, although tar itself is so cheap, so that it is extravagant to burn such oil as a fuel.

Experiments with shale-oil.

22) g. **Shale-oil** has been used by SELWYN in his experiments of late years. With the laudable object of producing the liquid fuel in England and thus keeping independent of foreign countries, SELWYN recommends that the heavy oil from Bog Head coal which occurs plentifully in Scotland, or from bituminous shale, of which oil 200000 tons are annually produced, should be used as a fuel in the British fleet. SELWYN gives the spec. gravity of this heavy oil as 1.050 to 1.060 but sometimes it rises to 1.075 and its flashing point is far above 100°, so that it is not at all dangerous. As it has a high calorific value, and perhaps could be produced in much larger quantities, SELWYN'S proposition would be worth attention if there were not already such an extensive demand for the oil for other purposes that it is worth 3 s. per ton for large orders. For the some reason a trial made by the American Bog Head Co. in 1885 had to be abandoned. Their steamer the "Himalaya" of 800 tons, trading on the Brazilian coast, was fired with heavy oil made there, where good steam coal cost 30 s. a ton. Even under these highly favourable conditions for the liquid fuel its cost was so high compared with coals that it did not pay, although considerably more power was got ont of the boiler with the liquid fuel than with coals.

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*) Annales de Chimie et de Physique. 1868. Vol. XV. P. 30.

**) Journal of the united service institution. 1885. P. 689.

23) **III. Gaseous fuels**, i. e. the inflammable gases disengaged from solid or liquid fuels, may be said to be at present scarcely used for marine boilers. Gaseous fuels may be made from combustible materials such as coal-dust, saw-dust, tan, &c. which can be burnt on ordinary grates only with difficulty or not at all, and are therefore nearly valueless. Gases possess besides the advantage of giving a perfect combustion with a comparatively small supply of air and are therefore very efficient fuels. (See further on, § 21, 57 on p. 218.) It is therefore probable that furnaces burning gas will the sooner displace the ordinary smoke and soot forming bar-grates, the better the former can be adapted to the requirements of an irregular and interrupted service. Object of their use.

24) h. **Gases from solid fuels** (wood, peat, lignite, coals) were burnt by MÜLLER and FICHET*) at Ivry under a land boiler set in masonry, with 32 % of economy over the use of the same solid fuels on a bar grate. The gases are produced in a separate furnace — *the generator* — by driving off the carburetted hydrogens of the solid fuel (generally coal) and gasifying the residue (coke); the gases are then led into the combustion chamber in connection with the boiler, after being mixed with hot air. In the generator the principal object is to convert the carbonic acid, formed by the fire at the bottom of a thick layer of fuel, into carbonic oxide on its upward passage through the coals (see § 20, 5), in fact to produce as much of the latter gas as possible, in order to convert it by combustion under the boiler into carbonic acid. According to F. FISCHER **) the average chemical composition of the generator gases is Origin and qualities.

Carbonic acid	5	to	6 %
„ oxide	21	„	24 „
Marsh gas	2	„	3 „
Hydrogen	6	„	7 „
Nitrogen	62	„	64 „

One kilo of coals produces about 4.5 cbm of generator gas at about 700° temperature.

25) i. **Gases from liquid fuels** were tried in marine boilers some years ago; from crude petroleum by FOOTE and from heavy tar-oil by DORSETT & BLYTHE, to which must be added the vapours of benzine and kerosine used recently by YARROW in his „*Zephyr*" launches. Experiments in this direction by other engineers have failed for want of practical adaptability. Kinds of the gases.

*) Wochenschrift des Vereines deutscher Ingenieure. 1881. P. 207.

**) Zeitschrift des Vereines deutscher Ingenieure. 1888. P. 18.

Vapour of crude petroleum.

26) *Foote's*) gas-furnace* was worked for several trial trips on board the United States Gunboat "Palos" in 1867. It consisted of a cast-iron retort, inserted in each furnace-tube, and into which crude petroleum was continuously pumped. A number of burners placed at the bottom of this vessel and fed from it, evaporated the petroleum. The vapour, mixed with compressed air and highly superheated steam, issued from another series of burners at the upper part of the retort and burnt with a clear and very hot flame. FOOTE claims to have evaporated in this manner 21 ℔s of water per pound of petroleum, which is of course a great exaggeration. The principal objection to his arrangement lay in the high temperature necessary for the complete evaporation of the crude oil, which heated the retorts so much as very soon to burn them through, besides the stopping up of the pipes leading to the burners with the incombustible residue of the oil. At the trials these pipes had to be cleared out after being at work 48 hours.

Vapour of tar-oil.

27) *Dorsett & Blythe's gas-furnace**)* tried on board the S. S. "Retriever" in 1868 consisted of two small vertical boilers serving as retorts, from which the tar-oil vapour was conducted at a pressure 3 to 3.5 atmospheres and about 500° temperature to the burners in the main boiler furnaces. On account of the high temperature of the oil vapour the retorts and vapour pipes had to be cleaded with a mixture of sand and fire-clay contained in a sheet iron casing. The evaporation attained was 12.35 ℔s of water per ℔ of oil, — not particularly satisfactory. This arrangement, like FOOTE'S, failed through the high temperature of the oil vapour and the rapid stopping-up of the pipes.

Petroleum vapour.

28) *In Yarrow's "Zephyr" launches***)* benzine is no longer burnt, as originally, on account of its great danger, but the less limpid and rather heavier refined petroleum of commerce, ordinarily used for lamps. As the flashing point of this oil lies between 25° and 30° it requires very careful attention in handling. YARROW forces the oil, by compressed air, from a completely closed vessel into a pipe placed in the boiler furnace. The oil is evaporated in this pipe and passes out in the gaseous state through a burner into the furnace. This method of firing is reported to have answered well in all the launches hitherto at work, probably because refined petroleum is easily evaporated and leaves no residue on burning.

*) American Artizan of May 8th. 1868.

**) Engineering II. 1868. PP. 324 and 340.

***) Engineering 1888. I. P. 518.

§ 20.

Combustion of fuels.

1) **I. The process of combustion.** The combination of a body with oxygen, accompanied by the appearance of flame, is called *combustion.* Combustion.

2) Every combustion is divided into two processes; during the first gases are liberated from the fuel, and during the second these gases are burnt together with the solid residue of the fuel (coke). The liberation of the gases from the fuel is effected by its becoming heated up to its ignition temperature in the furnace. Processes during combustion.

3) The principal combustible components of any fuel are carbon C, hydrogen H, and oxygen O, and of these, a certain portion of the hydrogen and oxygen is present in combination in the proportion of 1 to 8 by weight as H_2O or *water.* On the fuel being heated, this water passes off as steam, diminishing the heat of combustion by the amount of the latent heat required for its formation. At the high temperature of combustion the steam H_2O becomes afterwards decomposed, H combines with C to carburetted hydrogen, O with C to CO, *carbonic oxide*, and both these gases are by combustion with the air entering the furnace converted into CO_2, *anhydrous carbonic acid* generally called *carbonic acid* (which is really H_2CO_3), and H_2O or water. This part of the process however does not add to the heat of combustion, as, according to the law "*the quantity of heat which is liberated by the combination of two bodies becomes latent again at their decomposition*", only the same amount of heat can be gained by the combustion of the products of the decomposition of the aqueous vapour or steam as is just required to produce this decomposition. Formation and decomposition of aqueous vapour or steam.

4) If the fuel is heated further, its component parts which are richest in hydrogen are decomposed and every particle of the free hydrogen combines with three times its weight of carbon to form CH_4, light carburetted hydrogen *marsh-gas*, or with six times its weight of carbon, forming heavy carburetted hydrogen C_2H_4, *olefiant-gas*, both of which, being easily inflammable, are burnt under contact with the oxygen of the atmospheric air partly to CO_2 and partly to H_2O. Carburetted hydrogens.

5) After the liberation of the carburetted hydrogens, whose combustion raises the temperature very considerably, the other and less easily inflammable parts of the fuel (leaving a certain Carbonic oxide and carbonic acid.

portion of their carbon free in a finely divided state) combine with the oxygen O of the air, the nitrogen of which passes through the process uncombined. If, at this stage, insufficient air (or oxygen) is admitted, CO is formed and the combustion is incomplete, the carbon not having reached its highest grade of oxydisation, which is only attained with an ample air-supply. It has however no influence upon the temperature of combustion whether CO is first produced and then further burnt (or oxydised) to CO_2, or whether the carbon is burnt into CO_2 at once. On the other hand, CO_2 once having been formed, must not come into combination with hot carbon, because it will take up from this one equivalent of C and form $CO_2 + C = 2\,CO$ or in other words, become reduced to carbonic oxide, which would escape unburnt if it had no opportunity by contact with air in the furnace, of becoming again oxydised to CO_2.

Component parts of the products of combustion.

6) *Under perfect combustion of the fuel*, which is unattainable in marine boiler furnaces for the reasons detailed in § 21, 19, it is evident from the foregoing that the only gases escaping by the chimney will be *nitrogen*, as the incombustible portion of the air admitted to the fire, and *steam* and *carbonic acid*, the products of the combustion of the fuel.

100 parts, by weight, of	H_2O	consist of	11.11	parts of	H	and	88.88	parts	of	O
" " "	CO_2	"	" 27.27	"	C	"	72.73	"	"	O
" " "	CO	"	" 42.85	"	C	"	57.15	"	"	O

If any other gases than N, H_2O, and CO_2 leave the chimney, the combustion is imperfect. As these gases are more or less colourless, the escaping gases with perfect combustion are scarcely visible. If they are coloured light brown to deep black, i. e. if *smoke* issues from the chimney, this is an evidence of the more or less incomplete combustion which is going on, as the colouring matter of the smoke is carbon escaping unburnt. The absence of smoke, or the existence of only faint smoke, is however not always a proof of perfect combustion, for carbonic oxide is also colourless and if carbonic oxide alone instead of carbonic acid passes from the chimney, the fuel only develops $^1/_3$ of the heat produced under perfect combustion. The degree of completness of the combustion achieved with any fuel can only be determined by analysing the uptake gases — see 34 — *smoke analyses.*

Structure of luminous flame

7) *The smokeless combustion* of the solid residue (coke) of a fuel is attained without difficulty so long as the oxygen contained in the air-supply is sufficient to completely burn it, and the products of combustion are not cooled down below the ignition-

temperature of the carbon. According to KRÄMER*), the complete and smokeless combustion of gases giving a highly luminous flame is more difficult. This luminous flame consists of a core formed out of the unconsumed carbo-hydrides just liberated from the fuel and surrounded like a mantle by the incandescent products of combustion, carbonic acid and steam mixed with air. Between this mantle and the core is the luminous zone of the flame.

8) Within the luminous zone a decomposition of the carbo-hydrides takes place, the lower temperature of the mantle which surrounds the incandescent products of combustion causing carbon to be set free. At the interior portion of the luminous zone, only hydrogen is burnt, which by its heat of combustion produces the high temperature required to ignite the liberated carbon surrounding it and, by the incandescence of the latter, causes the luminosity of the flame. But if the carbo-hydrides can be mixed with sufficient air to bring about their immediate combustion, the formation of luminous flame, i. e. the separate combustion of the carbon and the hydrogen, may be avoided and their total calorific power, i. e. the sum of their separate heats of combustion, be made available. **Combustion in a luminous flame.**

9) The foregoing shews that when any fuel burns with luminous flame, only so much of the heat of combustion of the carbo-hydrides is fully developed as is due to the hydrogen, whereas only that portion of heat of combustion of the carbon is made available, which is developed by the free carbon in the luminous zone, minus the amount converted into light. By a good distribution of the air-supply, the formation of luminous flame and its attendant losses of heat may be to a certain extent reduced, but never so effectively with solid and liquid fuels as with gaseous ones. A BUNSEN burner fed with ordinary coal gas and supplied with sufficient air, giving a non-luminous flame is an example of complete combustion. Perfect combustion of this sort can however only be advantageously used in cases where a comparatively small surface is to be heated by *conduction*, or direct contact with the flame. For heating large spaces like boiler furnaces however, it is more profitable, according to FR. SIEMENS (see § 21, 6), not to attempt to reach this theoretically perfect combustion, but to produce a large luminous flame, which does its work by *radiation*. **Loss of heat attendant upon luminous flame.**

*) Sitzungsberichte des Vereins zur Beförderung des Gewerbfleisses. Berlin 1887. P. 104.

Division.

10) **II. Calorific and evaporative power.** We distinguish:

a) the absolute calorific power e_a,
b) „ useful „ „ e_n,
c) „ specific „ „ e_s,
d) „ pyrometric „ „ t_p,
e) „ theoretical evaporative „ e_v,
f) „ useful „ „ e_{v_n}.

Methods of determination.

11) a. **The absolute calorific power or heat of combustion** of a fuel is the number of thermal units developed when 1 kilo of the fuel is burnt. It may be determined either by

α) analysis and calculation,
β) or by direct calorimetric experiment.

Results of Favre and Silbermann's experiments.

12) α. *The determination of the absolute calorific power by analysis* is based on the results of FAVRE and SILBERMANN'S*) extensive calorimetrical experiments. They, and others after them, found the calorific power or thermal units per kilo completely burnt in pure oxygen to be for

C (charcoal) burnt to	CO	2473	T. U.
CO . . . . . „ „	CO_2	2403	„
C (charcoal) „ „	CO_2	8080	„
H . . . . . . „ „	H_2O	34462	„
CH_4	$CO_2 + 2\,H_2O$	13346	„
C_2H_4 . . .	$2\,CO_2 + 2\,H_2O$	11957	„
P (phosphorous) . . .	P_2O_5	5933	„
S (sulphur)	SO_2	2160	„
S	SO_3	2870	„

The absolute calorific power of those bodies which are burnt to H_2O is referred to liquid water at 0°. But as this water in reality escapes from the furnace in the form of steam, we must subtract from the foregoing figures the total heat of steam of 100° = *637* T. U. × the number of equivalents of water in the respective compound. For hydrogen, for instance $9 \times 637 = 5733$, and we get the absolute calorific power $= 34462 - 5733 = 28729$ T. U. In calculating the losses of heat occasioned by the furnace gases passing off (see § 21, 47), it is convenient to refer these gases (and the steam forming part of them) to the stokehole temperature of say 20°. Then 1 kilo of steam contains $637 - 80 \times 0.4805 = 598.66$, or in round numbers *600* T. U., and we get for one kilo of

H burnt to	H_2O . . . . about	29000	T. U.
CH_4 „ „	$CO_2 + 2\,H_2O$ „	11900	„
C_2H_4 „ „	$2\,CO_2 + 2\,H_2O$ „	11200	„

*) Annales de chimie et de physique. Sér. III. Tome XXXIV.

Finally the calorific power of sulphur, which occurs in fuels partly as pyrites, partly in organic compounds, and is therefore burnt in the former case to sulphurous acid SO_2, and in the latter to anhydrous sulphuric acid SO_3, — is generally taken at *2500* T. U. the mean, in round numbers, of the values for these two compounds.

13) According to DULONG'S rule "*the absolute calorific power e_a of a fuel is equal to the sum of the calorific powers of its constituents,*" Dulong's rule. if the absolute calorific power of a fuel is required, its chemical composition must first be determined by analysis and the sum of the calorific powers of its elements, as given above, obtained. The absolute calorific power of 1 kilo of a fuel, consisting only of C kilos of carbon, H kilos of hydrogen and O kilos of oxygen, would accordingly be calculated as follows:

C kilos of carbon burnt to CO_2 produce $8080\,C$ T. U.;

the free hydrogen left after the loss through forming water $= H - \frac{O}{8}$ kilos and this, when burnt to H_2O produces $34462 \times \left(H - \frac{O}{8}\right)$ T. U.; so that one kilo of this fuel developes, when completely burnt,

$$e_a = 8080\,C + 34462\left(H - \frac{O}{8}\right) \text{ T. U.} \quad \ldots \ldots \ldots \quad (92)$$

14) Later researches of HESS as well as FAVRE and SILBERMANN*) Dulong's rule improved. have shewn that DULONG'S rule is only correct for the combustion of elements and not of their compounds, but that the absolute calorific power of chemical combinations as met with in fuels, is equal to the sum of the heat quantities developed at each of the various chemical stages of their formation. Any quantities of heat lost during such processes are to be regarded as negative. We have for instance, in the case of

		observed absolute calorific power	sum of the heats of combustion of the elements	difference
Marsh-gas	CH_4	*13063*	*14607*	— *1612* T.U.
Olefiant-gas	C_2H_4	*11857.8*	*11848.8*	+ *9* „ ,

shewing that during the formation of marsh-gas from its elements *1612* T. U. are consumed, and with olefiant gas *9* T. U. are developed. So that for fuels which in burning produce carbo-hydrides, DULONG'S rule only gives the absolute calorific power approximately, on which account, in applying this rule the later averaged values are often used instead of the former exact figures of FAVRE and SILBERMANN mentioned in 12).

*) Annales de chimie et de physique. Sér. III. Vol. XXXIV.

Application of Dulong's rule in practice.

15) *The practical determination of the absolute calorific power of a fuel*, as proposed by the Verein Deutscher Ingenieure*) and the association of boiler inspection companies, is carried out as follows. From every load (truck or basket) of the fuel, one shovelful is taken and placed in a box. From time to time an average sample is taken of the contents of the box by breaking them up small, spreading them out in the form of a square on the floor, and dividing this square into four equal parts by two diagonals. Two of these parts are removed and the other two well mixed, broken up still more finely, and again divided in a similar manner in a square as before. This process is continued until a sample weighing about 5 kilos is obtained which is sealed up for chemical analysis. This will not be gone into here as it is a matter for a professional chemist. It embraces the investigation of the behaviour of the fuel under heat as well as its proportions of carbon, hydrogen, nitrogen, sulphur, ash, and hygroscopic water. For this last purpose a number of small samples should be taken and enclosed in sealed bottles. Assuming 1 kilo of the fuel to contain

C kilos of carbon,
H „ „ hydrogen,
S „ „ sulphur,
O „ „ oxygen,
W „ „ hygroscopic water,
N „ „ nitrogen,
and A „ „ ash;

its absolute calorific power e_a is calculated by DULONG'S formula, substituting the averaged values given in 12) as explained in 14), either from

$$e_a = 8100\,C + 29000\left(H - \frac{O}{8}\right) + 2500\,S - 600 \text{ T. U.}, \quad \ldots (92^a)$$

or from

$$e_a = 8000\,C + 29000\left(H - \frac{O}{8}\right) + 2500\,S - 600 \text{ T. U.} \quad \ldots (92^b)$$

For coals of average quality, the composition of which is given in § 19, 13, i. e. for $C = 0.8$ kg, $H = 0.04$ kg, $S = 0.02$ kg, $O = 0.08$ kg, $W = 0.03$ kg, we get by Eq. 92^a

$$e_a = 8100 \times 0.80 + 29000\left(0.04 - \frac{0.08}{8}\right) + 2500 \times 0.02 - 600 \times 0.03 = 7382 \text{ T. U.}$$

and similarly, by Eq. 92^b, $e_a = 7302$ T. U.

*) Zeitschrift des Vereines deutscher Ingenieure. 1884. P. 860.

16) *β. The determination of the absolute heating power by direct calorimetrical experiment* is, for scientific investigations, quite indispensable, as according to F. FISCHER*) DULONG'S formula is in fact inapplicable to coals. F. FISCHER found the absolute calorific power of a certain kind of coal to be **Exact determination of the absolute calorific power e_a.**

by calorimetrical experiment . . . . . *7720* T. U.
by DULONG'S formula . . . . *7175* „
Difference *545* „

Calorimeters suitable for testing fuels and the methods to be observed are described at length in the very excellent work referred to below.**)

17) b. **The useful calorific power** e_n of a fuel is the number of thermal units which 1 kg of it, burnt in a furnace, will give up to the water in the boiler. The value of e_n is obtained by measuring the water D evaporated and the fuel K burnt in a certain time. If the temperature of the feed-water is $t_1{}^0$, every kg of it must receive **Useful calorific power e_n of the fuel in its original state.**

$$\lambda - t_1 \text{ T. U.,}$$

before it can escape as steam. The value of λ is taken for the absolute boiler pressure from the table on pp. 28 to 31. The total heat-quantity taken up by the boiler water is therefore

$$D(\lambda - t_1) \text{ T. U.,}$$

so that the fuel has given up, per kg

$$e_n = \frac{D(\lambda - t_1)}{K} \text{ T. U.} \quad \ldots \ldots \ldots \ldots \quad (93)$$

which is *the useful calorific power* or *calorific value of the fuel in its natural state.* It varies for coals between 4500 and about 5500 T. U. according to their composition, the efficiency of the furnace, &c. Compare § 21, 61.

18) In scientific investigations it is usual to calculate the useful calorific power of the fuel when purified, i. e. dried and deprived of its ash. For instance, if in an experiment K kg of a fuel have been burnt, the analysis of which shewed W kg of water and A kg of ash per kilo of the fuel, then only **Useful calorific power e_n of the pure fuel.**

$$K - K(A + W) \text{ kg}$$

of the pure fuel are burnt, and accordingly *the useful calorific power of the pure fuel* is

$$e_n = \frac{D(\lambda - t_1)}{K - K(A + W)} \text{ T. U.} \quad \ldots \ldots \ldots \ldots \quad (93^a)$$

*) Zeitschrift des Vereines deutscher Ingenieure. 1886. P. 45.

**) F. FISCHER. Chemische Technologie der Brennstoffe. Braunschweig. 1882. P. 150.

In this way exacter figures are obtained for comparison, as the proportion of water and ash, which vary so considerably with the nature of the fuel, is left entirely out of account.

Determination of the specific calorific power.

19) c. **The specific calorific power** e_s of a fuel is the number of thermal units produced by the combustion of one litre of it and is equal to the absolute calorific power e_a multiplied by the specific gravity γ of the fuel

$$e_s = \gamma\, e_a \text{ T. U.} \qquad (94)$$

This expression is only used with reference to gaseous fuels, when it is known how many litres of gas are produced from 1 kg of the solid or liquid fuels from which they are generated. F. FISCHER*) gives the specific calorific powers of the following as

Gas		referred to liquid water at 0°	referred to steam at 20°
Benzole	C_6H_6	$e_s = 35.40$ T. U.	34.00 T. U.
Propylene	C_3H_6	$e_s = 22.50$ „	21.00 „
Olefiant gas	C_2H_4	$e_s = 15.00$ „	14.00 „
Marsh gas	CH_4	$e_s = 9.50$ „	8.54 „
Hydrogen	H_2	$e_s = 3.07$ „	2.58 „
Carbonic oxide	CO	$e_s = 3.05$ „	3.05 „

Determination of the pyrometrical calorific power.

20) d. **The pyrometrical calorific power or the temperature of combustion** t_p of a fuel is the temperature C, in degrees centigrade, produced by the combustion of 1 kg of it. This temperature can only be arrived at by calculation, as pyrometers are not sufficiently accurate for measuring combustion temperatures. The pyrometrical calorific power of a fuel equals the absolute calorific power divided by the product of the weight of gas produced during the combustion and its specific heat. As the specific heats for the gases of combustion at high temperatures are not accurately known (see § 5, 6), the pyrometrial calorific power can only be approximated to, and therefore the value of the following calculation is comparatively slight.

Formula for the pyrometrical calorific power.

21) If in 1 kg of a fuel

C kg of carbon burn to carbonic acid,
C_1 „ „ „ „ „ „ oxide,
H „ = the proportion of hydrogen,
W „ = „ „ „ hygroscopic water,
A „ = „ „ „ ash,
e T. U. = the absolute calorific power of the carbon burnt to carbonic acid,

*) Zeitschrift des Vereines deutscher Ingenieure. 1888. P. 17.

e_1 T. U. = the absolute calorific power of the carbon burnt to carbonic oxide,
e_2 T. U. = „ „ „ „ „ the hydrogen,
N kg = „ weight of nitrogen contained in the air-supply,
c, c_1, c_2, c_3, and c_4 = the respective specific heats of the carbonic acid, carbonic oxide, steam, and ash (see table on p. 8),
$r = 540$ T. U. the latent heat of steam;

then, assuming that the air-supply is only just sufficient for the combustion, that the initial temperature of all the bodies is 0^0, and the atmospheric pressure is constant at 76 cm of mercury, the pyrometrical calorific power is

$$t_p = \frac{e\,C + e_1\,C_1 + (e_2\,H - 9\,r\,H) - 540\,W}{3.67\,c\,C + 2.33\,c_1\,C_1 + 9\,c_2\,H + c_2\,W + c_3\,N + c_4\,A} \text{ degrees centigrade} \quad (95)$$

The numerical values 3.67, 2.33, and 9, in the denominator of this expression follow from 24), in accordance with which also $N = (2.67\,C + 1.33\,C_1 + 8\,H)\,3.19$, while $C + C_1 + H + W + A = 1$ by the hypothesis. If the combustion is complete, i. e. if the whole of the carbon is burnt to carbonic acid then $C_1 = 0$ and Eq. 95 becomes, after substituting the above values,

$$t_p = \frac{2785\,C + 10200\,H - 226\,W}{C + 0.16\,W + 3.65\,H + 0.07\,A} \text{ degrees centigrade} \quad (95^a)$$

It is calculated that the pyrometrical calorific power of

1 kg C	burning to	CO_2:	in oxygen $= 10200^0$,	in air $= 2730^0$
1 „ CO	„ „	CO_2:	„ „ $= 7180^0$,	„ „ $= 3040^0$
1 „ H	„ „	H_2O:	„ „ $= 6670^0$,	„ „ $= 2665^0$
1 „ CH_4	„ „	$CO_2 + 2\,H_2O$:	„ „ $= 7160^0$,	„ „ $= 2440^0$
1 „ C_2H_4	„ „	$2\,CO_2 + 2\,H_2O$:	„ „ $= 8620^0$,	„ „ $= 2750^0$

The pyrometrical calorific power is in reality less than that calculated by Eq. 95^a, as it is reduced by the dissociation of the products of combustion.

22) e. **The theoretical evaporative power** e_v of a fuel is the number of kilogrammes of water at 0^0 converted into steam at 100^0 by the combustion of 1 kg of the fuel. As the heat λ contained in the steam at 100^0 evaporated from water at 0^0 is 637 T. U., and one kg of the fuel developes e_a T. U., it follows that the theoretical evaporative power of the fuel is

Determination of the theoretical evaporative power.

$$e_v = \frac{e_a}{637} \text{ kg of water} \quad \ldots\ldots\ldots \quad (96)$$

Therefore, theoretically, 1 kg of coals evaporates on an average (referring to the values of e_a found by Eq. 92^a and 92^b) $e_v = \frac{7300}{637}$ to $\frac{7400}{637} = 11.4$ to 11.6 kg of water. — In England

the theoretical evaporative power is referred to water at 100° which is to be converted into steam at 100°, the heat of evaporation of which is only 536.5 T. U.; we then have for coals of average quality

$$e_v = \frac{7300}{536.5} \text{ to } \frac{7400}{536.5} \text{ in round numbers } 13.5 \text{ to } 14.0 \text{ kg.}$$

For the best descriptions of coal, the value of e_v may rise nearly to 15 kg.

Useful evaporative power e_{v_n}.

23) f. **The useful evaporative power** e_{v_n} of a fuel is the number of kilos of water of 0° temperature converted by the combustion of 1 kilo of the fuel in a boiler furnace, into steam of 100°. It is found by dividing the useful calorific power e_n of the fuel by 637, the total heat of 1 kg of steam at 100° produced from water at 0°. In the same manner as the useful calorific power both for crude (containing ash and water), and pure fuel is determined by Eq. 93 and 93ᵃ respectively, we can calculate the useful evaporative power

$$e_{v_n} = \frac{D(\lambda - t_1)}{637\,K} \text{ kg of water} \quad \ldots \quad (97)$$

for 1 kg of crude fuel, and

$$e_{v_n} = \frac{D(\lambda - t_1)}{637\,[K - K(A + W)]} \text{ kg of water} \quad \ldots \quad (97^a)$$

for 1 kg of pure fuel. If the useful evaporative power "from and at" 100° is required, we must substitute the number 536.5 instead of 637 in the above equations, as explained in 22).

Determination of the weight of oxygen.

24) **III. Air-supply. The weight of oxygen** necessary for the combustion of 1 kg of fuel composed according to 15) of C, H, O, S, &c. is computed as follows. C kg of carbon burnt to CO_2 which consists of $12\,C + 32\,O$, require $O_1 = \frac{32}{12} C = 2.67\,C$ kg of oxygen; $H - \frac{O}{8}$ kg of hydrogen, to form H_2O, containing $2\,H + 16\,O$, require

$$O_2 = \frac{16}{2}\left(H - \frac{O}{8}\right) = 8\,H - O \text{ kg;}$$

S kg of sulphur to form SO_3, consisting of $32\,S + 32\,O$, require

$$O_3 = \frac{32}{32} S = S \text{ kg;}$$

∴ 1 kg of fuel requires altogether

$$O = O_1 + O_2 + O_3 = 2.67\,C + 8\,H - O + S \text{ kg} \quad \ldots \quad (98)$$

As 1 kg of oxygen at 0° and 760 mm barometric pressure measures 1.43 cbm, the volume of the above weight of oxygen is

$$O' = \frac{2.67\,C + 8\,H + S - O}{1.43} \text{ cbm} \quad \ldots\ldots\ldots \quad (98^a)$$

25) **The quantity of air** which contains this amount of oxygen is **Theoretical air-supply.**

$$L = \frac{2.67\,C + 8\,H + S - O}{0.23} \text{ kg of air,} \quad \ldots\ldots \quad (99)$$

as 1 kg of the atmosphere consists of 0.23 kg of oxygen and 0.77 kg of nitrogen, in round numbers, — or in volumes

$$L' = \frac{2.67\,C + 8\,H + S - O}{0.21 \times 1.43} \text{ cbm,} \quad \ldots\ldots\ldots \quad (99^a)$$

as a cbm of air contains about 0.21 cbm of oxygen and 0.79 cbm of nitrogen. For converting cbm into kg of air, it may be noted that 1 cbm of air, at 0^0 and 760 mm barometric pressure, weighs 1.29 kg.

26) A kg of coals of the average analysis given in § 19, 13 would, by the above, require **Example.**

$$L = \frac{2.67 \times 0.80 + 8 \times 0.04 + 0.02 - 0.08}{0.23} = 10.42 \text{ kg or}$$

$$L' = \frac{10.42}{1.29} = 8 \text{ cbm of air.}$$

27) *For boiler furnaces with natural draught*, experience has shewn that the weight of air given by Eq. 99 is insufficient. It has in general been assumed that the actual air supply should be double the theoretical one. Recent investigations have however shewn that twice the theoretical air supply is in most cases for too great and that losses of heat always accompany an excess of air. According to these experiments, the excess of air above the theoretical quantity should be in proportion to the gaseous portion only of the products of combustion of the fuel. For coals, the most favourable combustion (see § 21, 58) is obtained (with natural draught) when the actual air supply is in the following proportion to the theoretical one **Actual air-supply required with natural draught.**

1.7 to 2.0 : 1, i. e. about 13 to 14 cbm or 16 to 18 kg for bituminous coals,
1.5 „ 1.7 : 1, „ „ 12 „ 13 „ „ 15 „ 17 „ „ semi-bituminous „ ,
1.3 „ 1.5 : 1, „ „ 11 „ 12 „ „ 14 „ 16 „ „ slightly-bituminous „ .

The method of calculating the excess of air from the analysis of the smoke is described in 38).

28) *With artificial draught* the excess of air above the quantity theoretically required may be reduced, — in favourable circumstances, almost to nothing. In such cases perfect combustion is obtained with the theoretical air-supply alone, as shewn by the following results of some trials made in 1888 by the Compagnie des Forges et Chantiers of Marseilles*) with a **Air-supply with artificial draught.**

*) A. BIENAYMÉ. Les machines marines. Paris. 1887. P. 469.

boiler, belonging to the Iron-clad "Marceau" 10581 tons, temporarily erected on shore and worked with forced draught:

Coals burnt per □ m of grate per hour kg	Air-supply per kg of coals consumed in cbm
101,30	12,410
150,70	12,315
201,45	11,060
250,00	11,220
300,00	8,500
300,00	8,500

The circumstance, that with forced draught the volume of blast can be lessened the more active the combustion becomes, is explained by the air being more intimately mixed with the gases of combustion as the blast pressure is raised in consequence of finding its way better through the interstices of the fuel, so that the combustion approaches the process described in 8).

Air-supply.

29) **The air-supply,** i. e. not only the weight of air supplied to the fire, but the method of conducting it into the furnace, has a very great influence upon the temperature of the gases and therefore upon the attainable degree of perfection of combustion. The best method of distributing the air-supply will be further gone into under the heading of furnaces. For the present the influence of the quantity only of the air will be briefly discussed, with especial reference to solid fuels. In general, four cases may be distinguished:

a) quantity and diffusion of the air sufficient,
b) quantity sufficient, diffusion defective,
c) quantity deficient,
d) quantity excessive.

Visible signs of this condition.

30) a. *The quantity and diffusion of the air are sufficient* (for solid fuel) only when the fires have burnt through, whereas on the doors being opened the air supply is always too great, and when fresh fuel is thrown on, the air cannot mix sufficiently with the gases suddenly evolved. The fuel is burnt as completely as possible when these two, quantity and distribution or diffusion, are kept just within the most favourable limits. The evidences of this state of affairs are brilliantly illuminated ash-pits and only very slight smoke.

Symptoms.

31) b. *The quantity being sufficient, the diffusion is defective*, when portions of the grate are left bare, through the openings of which a portion of the air can enter the furnace and escape without permeating the fuel. Part of the carbo-hydrides then remain unconsumed, and possibly some carbonic acid, once

formed, may have an opportunity of becoming again reduced to carbonic oxide through contact with incandescent carbon, see 5). There is no external symptom of these unsatisfactory conditions unless unburnt carbon issues from the funnel as smoke, or the boilers hum in consequence of the violent draught passing through the bare portions of the grates.

32) c. *The quantity of air is always too small* when fresh fuel is thrown on, or when the sectional area of the air spaces through the grate is diminished by clinker, or when the velocity of the air, — which in marine boilers with natural draught is about 1.5 to 1.7 m per sec. at the ash-pit mouths, — is reduced by the influence of the weather. In all cases as much oxygen as can get at the fuel is used in forming steam and carbonic acid, while that part of the carbon which does not meet with oxygen sufficient to completely oxydize it, escapes in the form of free carbon or smoke, or else as carbonic oxide gas. As, consequently, only a limited part of the combustible components of the fuel come to complete combustion, its useful calorific power must fall, while on the other hand its pyrometrical calorific power rises, the heat of combustion being distributed over only a small volume of air, the temperature of which is thus unduly raised. As the current of air through the grate is weak and does not cool the bars sufficiently, the high temperature of the fire causes them to get out of shape, burn on their upper surface, and finally fall into the ash-pit. An unmistakeable sign of insufficient air-supply is increased radiation into the stokehole caused by the high temperature of the fire, which is oppresively perceptible as it is necessarily accompanied by a want of the proper movement of air towards the ash-pits. **Diminution of the calorific effect, rise of temperature, heat in the stokehole.**

33) d. *The air-supply is too great* in by far the most marine boilers, on account of unnecessarily large grate-surface. So great an excess of air passes through the spaces of the bars that not nearly all its oxygen is used, its nitrogen of course only serves to lower the temperature of the fire, and most of the heat produced is lost up the funnel. F. FISCHER*) quotes the case of a boiler examined by him in which the heat passing up the chimney was 60 % of the total calorific value of the coals. There is not only no symptom by which this wasteful state of affairs discovers itself, but the external signs of it do not differ from those of the most favourable combustion described in 30). The one and only infallible means of getting at the **Losses of heat with excessive air-supply.**

*) F. FISCHER. Taschenbuch für Feuerungstechniker. Stuttgart 1883. P. 29.

efficiency of boiler furnaces, of illustrating the state of affairs as to air-supply and losses of heat, is — the smoke analysis recommended in 6).

Object.

34) **IV. Smoke analysis,** which ought to be made part of the business of all trial trips, provides us, by means of determining the composition of the smoke and furnace gases, with the data for judging of the degree of completeness of the combustion and the only trustworthy basis for calculating the amount of heat carried off by these gases through the funnel and lost. Assuming, as we are entitled to do, that all fuels are about equally good on an average, we can by means of smoke analysis compare the economy and efficiency of the different types of marine boilers. The apparatus used, which will now be described, was devised by SCHLÖSING and ROLLAND and afterwards perfected by ORSAT.

Description and method of using.

35) *Orsat's apparatus* *), which is based upon the capacity possessed by certain fluids for absorbing certain gases, is designed to determine the proportion of carbonic acid, oxygen, and carbonic oxide the funnel gases contain. The apparatus consists of an aspirator, a graduated tube, three glass bottles with bell-shaped covers and each provided with a tube projecting downwards into a vessel partly filled with water to form a hydraulic joint. The aspirator serves to pass the gases into and through the apparatus. The three bottles are filled with three different fluids to absorb the different gases. The covers of the three bottles are connected to a tube which leads into the funnel and can be shut off by a cock. It is preferable to make this tube of glass, as iron tubes take up oxygen even at a low temperature and part with it again to reducing gases. The tube forming the hydraulic joint connected with each of the bottles is surrounded by an external vessel partially filled with water to protect the fluid enclosed in the bottle from the access of the air, the oxygen of which would soon decompose it. On the uptake gas being forced into these vessels, the water is displaced from them, and passing into the intermediate space between the two, forms a hydraulic joint. To use the apparatus, all the cocks are shut and air is blown by a pair of bellows through the tube connecting the bottles, the graduated tube, and the aspirator, to purify them. The graduated tube, and the aspirator being now filled with distilled water containing a few drops of hydrochloric acid, the cock leading to the

*) A. LEDIEU. Les nouvelles machines marines. Paris 1882. Vol. III. P. 383.

funnel is opened and the funnel gases sucked by means of the aspirator into the graduated tube until 100 parts of it are filled with gas. The connection with the funnel is now shut off, the cock of the first bottle opened and the gas forced into it from the graduated tube. The gas enters the fluid in the bottle by a number of glass tubes provided for the purpose of obtaining a large surface of contact. The fluid is a solution of potash of 1.26 to 1.28 sp. gravity which greedily absorbs the carbonic acid contained in the gas. When the gas has passed several times through the first bottle, it is sucked back into the graduated tube and the number of divisions it now fills read off. The difference between this reading and the original 100 parts has been absorbed as carbonic acid. The first bottle is now shut off, and the same process gone through with the second, which contains pyrogallate of potash (18 grammes of pyrogallic acid, 40 grammes of hot water and about 70 cbcm of the above potash solution) to absorb the uncombined oxygen. Finally the carbonic oxide is removed in the third bottle. This contains about 50 grammes of cuttings of sheet copper in 35 gr of chloride of copper with 200 cbcm of hydrochloric acid, to which after 24 hours 120 cbcm of water have been added. The residue of the uptake gas consists of nitrogen and small quantities of carbo-hydrides, the measurement of which is however, unimportant. The process is so simple that any attentive engineer can, with a little practice, make the analyses every four or five minutes, correct within 0.2 %. One charging of the bottles is sufficient for about two hundred analyses.

36) *In F. Fischer's*) small apparatus*, which in other respects is similar to ORSAT'S, the bottle for carbonic oxide is omitted because ordinary funnel gases contain but little of this compound. If a noticeable quantity of carbonic oxide is present, it is almost always accompanied by hydrogen and light carburetted hydrogen (marsh-gas CH_4). In order to determine these, a sample of the funnel gases must be sealed up in a glass tube and examined in the laboratory over mercury. The necessary apparatus is not further described here, as deeper investigations of this sort can only be undertaken by professional chemists. FISCHER'S apparatus is almost always sufficient for practical purposes and as its parts are kept as small as possible, it is only 40 cm high, 20 cm wide, and so light as to be very conveniently handled. It is to

Simplified apparatus.

*) F. FISCHER. Technologie der Brennstoffe. Braunschweig 1883. P. 221.

be wished that the low price of this useful apparatus, only 45 s., will lead to its becoming known and largely applied in practice. It is worked in the same manner as ORSAT'S. Fuller directions for the use and management of these apparatus will be found in F. FISCHER'S Technologie der Brennstoffe*).

Experiments in the German and French Navies.

37) Smoke analyses have been instituted in the German and French Navies, in order to determine the effect of artificial draught upon the process of combustion. In the German Navy Körting's blast was used and in the French, forced draught on the closed stoke-hole principle. In both cases the consumption of coal of course rose with increased draught or air pressure. In the French experiments the combustion became more and more complete as the intensity of the draught was augmented, but in the German ones, with Körting's blast, the combustion was less satisfactory, as may be seen from the table on p. 175.

Calculation of the excess of air.

38) The ratio v of the actual volume of air used vL', to the theoretical volume L', is obtained from the smoke analysis by the following method, where

k = the percentage volume of carbonic acid,
o = " " " " oxygen,
n = " " " " nitrogen.

The percentage volume o of uncombined oxygen escapes with the unused air, which always contains 21 volumes of oxygen to 79 volumes of nitrogen, so that its proportion of nitrogen is $\frac{79}{21}o$. Of the percentage volume of nitrogen n contained in the whole volume vL' of air used, there can only escape $n - \frac{79}{21}o$ as free nitrogen up the funnel, and only the quantity of air corresponding to this proportion of nitrogen was really required for combustion. So that we must have the relation

$$\frac{vL'}{L'} = \frac{n}{n - \frac{79}{21}o}$$

or, dividing by n and simplifying

$$v = \frac{21}{21 - 79\frac{o}{n}} \quad \ldots \ldots \ldots \ldots \ldots (100)$$

In the experiments recorded in the table on p. 175, the quantity of air used was 1.15 to 1.3 times the theoretical quantity. If this calculation is to be carried out with critical exactness, which is unusual, the proportion of oxygen in the air in the

*) Ibid. P. 244.

Table referring to smoke analyses with marine boilers.

Class and Name of Ship	Nature of the trial	Results of the analyses: Nitrogen N	Carbonic acid CO_2	Carbonic oxide CO	Uncombined oxygen O	Combined oxygen ÷ oxygen passing through grate $\frac{O_g - O}{O_g}$	Coal consumed per □ m of grate per hour in kilos	Temperature in funnel	Draught in funnel in mm of water	Height of barometer	Temperature of the air in degrees centigrade	Number of trials for which the figures given are mean values	Duration of the trials in hours
1	2	3	4	5	6	7	8	9	10	11	12	13	14
German Cruizer **"Albatross"**	Natural draught	83,16	10,62	2,87	3,35	0,848	92,2	215,0	4	766,70	8,00	6	6
" " "	Do.	83,40	11,96	2,24	2,40	0,892	101,8	248,0	6	753,25	7,25	5	4
" " "	Blast in funnel	83,95	7,80	4,20	4,05	0,790	106,6	272,5	55	754,00	7,25	2	2
French II Class Dispatch Boat **"Labourdonnais"**	Forced draught, closed stoke-hole	80,50	14,00	2,30	3,20	0,851	129,5	about 350,0	68—160 plenum of air in stoke-hole	—	—	—	—
French Iron-clad **"Marceau"**	Do.	80,15	14,25	0,45	5,15	0,763	201,5	—	—	—	—	—	—
"	Do.	80,05	13,85	0,70	5,40	0,748	250,0	—	—	—	—	—	—
"	Do.	80,10	15,60	0,40	3,90	0,816	300,0	—	—	—	—	—	—
"	Do.	80,10	16,60	0,70	2,60	0,880	300,0	—	—	—	—	—	—

stokehole, varying from 20.5 to 21 percentage volume, must first be tested with ORSAT'S apparatus. If we call the percentage volume of the oxygen thus found x and that of the nitrogen z, we have

$$v = \frac{x}{x - \frac{z}{n} o} \quad \ldots \ldots \ldots \ldots \quad (100^a)$$

Ratio of the volumes of oxygen.

39) *The ratio of the oxygen used O, to the oxygen O_g which passes through the grate* may also be taken as a measure of the completeness of the combustion, instead of the number denoting the excess of air. As air contains $\frac{79}{21} = 3.7619$ volumes of nitrogen to 1 of oxygen, if the analysis shews N volumes of nitrogen, then there have passed through the grate

$$\frac{N}{3.7619} = O_g$$

volumes of oxygen. Therefore $O_g - O$ volumes have been used, as O volumes have escaped. We thus get the ratio of oxygen used to that entering the grate as

$$\frac{O_g - O}{O_g} = \frac{\frac{N}{3.7619} - O}{\frac{N}{3.7619}} = 1 - 3.7619 \frac{O}{N}, \quad \ldots \ldots \quad (100^b)$$

from which formula col. 7 of the table on p. 175 is calculated.

Observations of the smoke on English trial-trips.

40) Smoke analyses taken on trial trips give a much firmer basis for judging of the efficiency of the furnace arrangements then can be obtained by simple personal observation as ordered to be made on English trial-trips*). According to the later English instructions of 1879, these observations are therefore to be omitted on trial-trips where Nixon's Navigation Coals are used. When these observations are to be made the degree of smoke is estimated by a person placed near the funnel. Eight numbers are employed in comparing the smoke, from 0 = *invisible* to 7 = *deep-black*. On measured-mile trials these numbers are noted at intervals of 1 minute, on extended trials at intervals of 5 minutes, — on a form supplied for the purpose. The sum of these numbers divided by the number of observations gives the average degree of smoke for the trial. The greatest degree of smoke occurring during 10 observations, as well as the time within which no smoke is observed, are also noted.

*) The steam trials of her majesty's ships. Engineering Oct. 15 and Dec. 31 1875.

41) **The gases of combustion** of fuels, when completely burnt in air, consist of Gases of combustion.

a) moist gases or steam,
b) dry gases, viz. carbonic acid, nitrogen, and sulphurous acid.

When the combustion is incomplete the latter gases may be accompanied by carbonic oxide, marsh-gas, hydrogen, and oxygen.

42) a. **The quantity of steam** W_r at 0^0 and 760 mm height of barometer contained in the gases of combustion of 1 kilo of a fuel is made up of Moist gases.

1) the proportion of water in the fuel, W kilos (see 15),
2) the water formed by the combustion of H kilos of hydrogen in the fuel $= 9\,H$ kilos,
3) the water suspended in the air which supports the combustion. To determine this, the simplest way is to ascertain the dew-point of the air with a hair hygrometer and then take the number of grammes of water f contained in 1 cbm of the air from the following table.

Table of moisture of the air.

Temperature	Water	Temperature	Water	Temperature	Water
1	2	1	2	1	2
1 °C.	5,2 gr.	11 °C.	10,0	21 °C.	18,2
2 "	5,6 "	12 "	10,6	22 "	19,3
3 "	6,0 "	13 "	11,3	23 "	20,4
4 "	6,4 "	14 "	12,0	24 "	21,5
5 "	6,8 "	15 "	12,8	25 "	22,9
6 "	7,3 "	16 "	13,6	26 "	24,2
7 "	7,7 "	17 "	14,5	27 "	25,6
8 "	8,1 "	18 "	15,1	28 "	27,0
9 "	8,8 "	19 "	16,2	29 "	28,6
10 "	9,4 "	20 "	17,2	30 "	30,1

As by Eq. 100, v times as much air was supplied to the fuel as is theoretically necessary and by Eq. 99ª, the theoretical quantity is L' cbm, we get the water due to the moisture of the air per kilo of fuel burnt

$$f \times v \times L' \text{ gr} = \frac{f \times v \times L'}{1000} \text{ kg}$$

and consequently

$$W_r = W + 9\,H + 0.001\,f\,v\,L' \text{ kg}, \quad \ldots \ldots \quad (101)$$

or as 1 cbm of this steam weighs 0.8048 kilos

$$W_r' = \frac{W + 9\,H + 0.001\,f\,v\,L'}{0.8048} \text{ cbm} \quad \ldots\ldots\ldots \quad (101^a)$$

If no very high degree of exactness is sought, the last term is neglected, particularly with dry air. Besides, absolutely correct measurements of the moisture cannot be obtained with a hair hygrometer. If however the last term is to be regarded, we may in most cases be satisfied with the approximate determination of $v \times L'$ given in 44).

Example.

43) For coal of average quality (§ 19, 13) the steam passing off in the gases of combustion per kilo of coal burnt is calculated as follows,

1) the $W = 0.03$ kilos of water give .... 0.33 kilos,
2) the $H = 0.04$ „ „ hydrogen form $9 \times 0.04 = 0.36$ „ ,
3) if the temperature of the air is 20^0 and it contains moisture, then by the table $f = 17.2$. Therefore for average coals, by Eq. 100 $v = 1.3$ as a mean, and by 26) $L' = 8$ cbm, then

$$0.001\,f\,v\,L = 0.001 \times 17.2 \times 1.3 \times 8 = 0.178 \text{ kg},$$

$$W_r = 0.03 + 0.36 + 0.178 = 0.568 \text{ kg},$$

$$\text{or} \quad W_r' = \frac{0.568}{0.8048} = 0.705 \text{ cbm of steam.}$$

Dry gases of combustion.

44) b. **The quantity of dry gases of combustion corresponding to 1 kilo of fuel** and referred to 0^0 temperature and 760 mm height of barometer is thus computed: if the fuel contains C kilos of carbon per kilo after deducting any unburnt carbon contained in the ash and clinker, this $1.854\,C$ forms K cbm of carbonic acid, and if there are besides, as given in 38) the percentage volumes k, o and n, then 1 kilo of the fuel forms

$$K\,o\,\frac{1}{k} = O_o \text{ cbm of oxygen and}$$

$$K\,n\,\frac{1}{k} = N_o \quad „ \quad „ \text{ nitrogen.}$$

Finally, the S kilos of sulphur which may be present produce $2\,S$ kilos or $\frac{2\,S}{2.864}$ cbm of sulphurous acid, as 1 cbm of it weighs 2.864 kilos. Consequently the dry products of combustion assume a volume V_r of

$$V_r' = K + O_o + N_o + \frac{2\,S}{2.864} \text{ cbm or}$$

$$V_r' = K + \frac{K\,(o + n)}{k} + \frac{2\,S}{2.864} \text{ cbm} \quad \ldots\ldots \quad (102)$$

As $K + O_0 + N_0$, except for the oxygen taken up in burning the hydrogen, almost represents the whole volume $v L'$ of the air-supply, we may put approximately (see 42)

$$v L' = K + O_0 + N_0$$

1 cbm	of carbonic acid	weighs	1.9781 kg,
1 „	„ oxygen	„	1.4303 „ ,
1 „	„ nitrogen	„	1.2566 „ ,

so that the weight of the dry gases of combustion is

$$V_r = 1.9781\, K + 1.4303\, O_0 + 1.2566\, N_0 + 2\, S \text{ kg}$$

and, expressing K in terms of C

$$1.9781\, K = 1.9781 \times 1.854\, C = 3.667\, C$$

$$V_r = 3.667\, C + 1.4303\, O_0 + 1.2566\, N_0 + 2\, S \text{ kg} \quad \ldots \quad (102^a)$$

45) If we assume that the coal of average quality (§ 19, 13) containing per kilo $C = 0.80$ kilos of carbon, $S = 0.02$ kilos of sulphur, &c., gives, on being burnt, a smoke analysis shewing Example.

$k = 15$ percentage volumes of carbonic acid,
$o = 5$ „ „ „ oxygen,
$n = 80$ „ „ „ nitrogen,

then these coals produce per kilo

Carbonic acid	1.854×0.80	$= 1.483$ cbm;	or $1.483 \times 1.9781 = 2.933$ kg
Oxygen	$1.483 \times 5 \times \frac{1}{15}$	$= 0.494$ „ ;	„ $0.494 \times 1.4303 = 0.277$ „
Nitrogen	$1.483 \times 80 \times \frac{1}{15}$	$= 7.909$ „ ;	„ $7.909 \times 1.2566 = 9.935$ „
Sulphurous acid	$\frac{2 \times 0.02}{2.864}$	$= 0.014$ „ ;	„ $0.014 \times 2.864 = 0.040$ „
Together	. . . .	9.900 cbm;	or . . . . 12.188 kg

of dry gases of combustion. The amount of sulphurous acid fumes is only given above, in order to shew that it is so small as not to require notice.

46) c. **The total quantity of gases of combustion produced from 1 kilo of fuel,** referred to 0° and 760 mm height of barometer, is composed of the dry gases determined by Eq. 102 and the watery vapour or steam (Eq. 101), and may therefore be expressed as Total gases of combustion.

$$\Phi = W_r + V_r \text{ kg or} \quad \ldots \ldots \ldots \quad (103)$$

$$\Phi' = W_r' + V_r' \text{ cbm} \quad \ldots \ldots \ldots \quad (103^a)$$

For the above example we get

$$\Phi = 12.188 + 0.568 = 12.756 \text{ kg},$$

$$\Phi' = 9.900 + 0.705 = 10.605 \text{ cbm}.$$

47) *The approximate volume V_r' of the dry gases* at 0° and 760 mm barometer, may be calculated from the k percentage volumes Approximate calculation of the dry gases.

of carbonic acid given by the smoke analysis and the proportion C kg of carbon contained in 1 kg of the fuel. The dry gases contain $\frac{k\,V'_r}{100}$ cbm of carbonic acid, which again contains 0.54 kg of carbon per cbm, so that altogether the gases carry off

$$\frac{0.54\,k\,V'_r}{100} \text{ kg of carbon.}$$

Upon the assumption that the C kg of carbon, shewn by analysis as contained in 1 kg of the fuel, are completely burnt to carbonic acid, the carbon in the gases must be equal to the carbon in the fuel, so that

$$C = \frac{0.54\,k\,V'_r}{100}$$

$$V'_r = \frac{100\,C}{0.54\,k} = 185\,\frac{C}{k} \text{ cbm} \quad \ldots\ldots\ldots \quad (103^b)$$

Approximate calculation of the total gases of combustion.

48) *The approximate quantity of the total gases produced from 1 kg of fuel*, referred to 0^0 and 760 mm barometer, is obtained by adding to V'_r the steam W'_r which, neglecting the last term of Eq. 101ᵃ, is

$$W'_r = \frac{W + 9\,H}{0.8048} = 1.24\,W + 11.18\,H \text{ cbm}$$

whence follows

$$\Phi' = 185\,\frac{C}{k} + 1.24\,W + 11.18\,H \text{ cbm} \quad \ldots\ldots \quad (103^c)$$

in which expression

C = the kilos of carbon per kilo of fuel,
H = „ „ „ hydrogen „ „ „ „ ,
W = „ „ „ hygroscopic water „ „ „ „ ,
k = „ percentage volume of carbonic acid in the gases.

Division.

49) **VI. The residue of combustion** of fuels in boiler furnaces arises from

a) imperfect combustion,
b) the incombustible parts of the fuel.

Residue from imperfect combustion.

50) a. **The residue consequent upon imperfect combustion** is either solid, as *coke* and *soot*, or gaseous as *carbonic oxide*, &c.

Coke.

51) *Coke* is coal which is unburnt though deprived of its volatile components, and is produced by insufficient access of air in the high temperature of the furnace. Its colour is blackish grey to iron grey and it has a faint metallic lustre. By this latter quality the small pieces of coke which fall into the ash-pit may be clearly distinguished from the other solid residue. As the calorific power of coke is about 7000 to 7500 T. U. it

is worth while wherever great economy of fuel is of importance, for instance in fast cruizers, to have the fragments of coke picked out of the ashes and put back in the fires from time to time.

52) *Soot* is unburnt carbon which may be formed by a too low temperature of the fire induced by excessive air supply, particularly when the fire-doors are opened. **Soot.**

53) *Carbonic oxide, hydrogen, marsh-gas, and sulphur fumes* pass off among the gases of combustion when the oxygen necessary to burn them to carbonic acid, steam, and sulphurous acid respectively, is either not supplied in sufficient quantity with the air, or the latter is not duly mixed with the liberated gases. **Unburnt gases.**

54) b. **The residue from the impurities of the fuel** is called by the generic name of *ash.* It contains the incombustible parts of the fuel which, according to the nature of the latter, may consist of silicates of alumina or of sulphate of lime and sulphide of iron, and may also contain lime, magnesia, oxides of iron and manganese, minute quantities of chlorine, traces of iodine, &c. According to their composition ashes may be classified as **Ash.**

a) Powdery, non-fusible ash, rich in alumina and containing little silica.
b) Friable unmelted ash consisting chiefly of silica without oxide of iron. Both these kinds of ash are from a greyish white to dirty grey in appearance.
c) Melted ash, or *clinker*, formed of oxide of iron and potassa silicates. Clinker resembles iron in colour and has a metallic lustre.

It may be taken as a rule in practice that clayey and siliceous ashes remain in a powdery state and therefore hinder combustion but little. The presence of iron and lime in the fuel causes the formation of clinker which on the one hand retards combustion by obstructing the air-spaces of the grate, and on the other hand reduces the calorific effect by fragments of coal becoming enclosed or embedded in the clinker.

55) The following table contains only average values for some of the best-known fuels, because the composition of any one kind of wood, coal, or oil, varies so much that exact figures can only be given specially for each description by itself. In using the table the following points are to be noted. **Remarks on the table.**

1) In columns 3—5 the chemical composition is given in percentage-weights of the absolutely pure fuel, i. e. after deducting the water and ash always present in the crude state.

Table of fuels.

Consecutive number	Description of Fuel	Components: Average composition of the chemically pure fuel in percentage weights: Carbon	Hydrogen	Oxygen	Average percentage weight in the crude, air-dried condition, of: Water	Ash	Calorific value: Average absolute calorific power in T. U.	Average pyrometrical calorific power in Degrees C.	Average useful evaporative power kg	Gases of combustion: Theoretical quantity of air necessary for combustion per kg in cbm	Gases produced by the combustion per kg at 0 °C in cbm	300 °C in cbm	Weight and stowage space: Average specific gravity	Average weight per cbm including interstices kg	Bunker space per 1000 kg cbm
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
	I. Solid fuels.														
1	Leafwood	48,5	6,2	45,3	17	1,0	2800	1900	2,5	3,5	4,2	8,8	0,65	420	2,60
2	Pine	50,2	6,1	43,7	15	0,5	3600	2200	3,5	3,8	4,5	9,5	0,50	380	3,00
3	Turf	55,2	6,0	38,8	30	5,0	2500	1800	1,5	4,0	4,6	9,7	0,38	250	4,00
4	Peat	60,5	5,9	33,6	25	8,0	3500	2100	3,0	4,2	4,8	10,0	0,50	450	2,30
5	Fibrous lignite	69,8	5,9	24,3	20	7,5	3600	2200	2,0	4,3	5,0	10,5	0,90	600	2,00
6	Earthy „	74,2	5,8	21,0	20	10,0	4500	2300	4,5	4,8	5,5	11,5	1,10	700	1,50
7	Younger coal	76,2	5,6	18,2	5	5,0	6500	2600	7,0	7,0	7,4	16,0	1,25	750	1,30
8	Older „	90,5	5,0	4,5	5	3,0	7800	2700	8,0	8,0	8,4	18,0	1,30	900	1,25
9	Pennsylvanian anthracite . . .	94,2	2,5	3,3	3	2,0	8000	2750	9,0	8,5	8,7	18,3	1,50	1000	1,20
10	Isère „ . . .	96,8	1,5	1,7	3	4,0	8200	2800	10,0	8,6	8,8	18,5	1,70	1100	1,15
	II. Liquid fuels.														
1	Pennsylvanian earth-oil . . .	84,9	13,7	1,4	2	0,0	11500	2800	15,0	11,2	11,8	25,0	0,88	880	1,10
2	Light Caucasian „ . . .	86,3	13,6	0,1	2	0,0	12200	2800	16,0	11,3	11,9	25,0	0,88	880	1,10
3	Heavy „ „ . . .	86,6	12,3	1,1	2	0,0	11200	2800	15,0	11,0	11,9	25,0	0,94	940	1,07
4	Astatki	87,1	11,7	1,2	2	0,0	11000	2780	14,0	10,8	11,8	25,0	0,93	940	1,08
5	Crude coal tar	87,2	5,3	7,5	8	0,0	8600	2700	10,0	9,2	11,2	23,5	1,20	1200	0,85
6	Scotch shale-oil	87,0	11,7	1,3	2	0,0	11000	2750	14,0	10,8	11,8	25,0	0,95	950	1,05

2) Column 6 gives the hygroscopic water usually retained by the fuel in its air-dried condition.
3) Column 7 gives the mean proportion of ash, i. e. of incombustible components, which however, is not to be confounded with the ordinary residue of combustion in marine boiler furnaces. The percentage of ash in a sample of fuel is determined in the laboratory by calcining about 5 grammes of it in a platinum crucible.
4) In columns 8 to 10 the mean calorific value is shewn in round numbers, and in particular, the "useful evaporative power" signifies the number of kilos of *water at 0°* converted by 1 kilo of the fuel as burnt in an ordinary boiler furnace, into *steam at 100°*.
5) Columns 11 to 13 contain the theoretical quantity of air for the combustion of 1 kilo, *without any excess*, and the volume of gases arising from this combustion, taken at 0° and at 300°, with the barometer at 760 mm. The pyrometrical calorific power in col. 9 is calculated upon the assumption that only the theoretical quantity of air is supplied to the fire. Of course if this air-supply is exceeded, the values in col. 9 will be proportionately reduced.
6) In col. 15 the weight of a cbm of the fuel is given, taking the interstices into account, and in col. 16, the space shewn by experience to be required for 1000 kilos in bunkers of the ordinary, often curved, forms.

§ 21.

Losses of heat in the combustion of fuels.

1) **I. Imperfect arrangement of the furnace** and inefficient firing condition the magnitude of the losses of heat which are specially due to Different kinds of losses of heat.
   a) defective combustion of the fuel,
   b) high temperature of the residue or ash,
   c) radiation and conduction to the outside.
2) A furnace-arrangement will be so much the more efficient, Characteristics of a good furnace.
   α) the better the space for combustion is adapted to the peculiarities of the fuel,
   β) the more nonconducting the walls of this space are,
   γ) the more easily the air-supply can be regulated.

Influence of the height (or diameter) of the furnace.

3) a. **The height of the space for combustion,** the base-area of which is a function of the quantity of fuel to be burnt in unit time, depends upon whether the fuel forms a short or long flame. This height must not be so small as to detract from the completeness of the combustion, nor so great as to diminish the effect of the radiation.

Advantages of roomy furnaces.

4) *In a furnace of sufficient height (i. e. diameter)* the flames have liberty to develope themselves without coming into contact with the furnace walls; the particles of carbon are then consumed before they leave the zone of combustion (see § 20, 8) and pass into the "mantle" consisting of the *products* of combustion (in which case these particles issue from the funnel as smoke). The resulting complete combustion produces a high temperature and therefore strong radiation. In a roomy furnace also the disproportion, between the great quantity of gases, suddenly disengaged from each supply of fresh fuel, and the (practically) constant volume of air supplied through the grate, is less marked than in a narrow furnace in which the furnace walls penetrate the "mantle" of the flame, thus hindering the free access of air to, and its due intermixture with, the gases.

Disadvantages of narrow furnaces.

5) *In low furnaces* the zone of combustion is elongated and (partially at any rate) removed from the place where the air enters, which retards the combustion, and cools the mantle of the flame by increasing its surface. This cooling action causes loss of heat at once, by the liberation of carbon from the carburetted hydrogen gases, for this carbon must either pass off as smoke, or be deposited upon the tubes as soot, because it can never again, in its further course towards the funnel, meet with a sufficiently high temperature to burn it. On the other hand the lower temperature of the fire reduces the radiation.

Economical use of radiation.

6) *The radiation* is consequently so much the more effective, the more completely the fuel is burnt, the more *compact* the flame, and the higher its temperature. The destruction of the furnace walls when only exposed to radiation is much less rapid than when they experience the direct action of the flame, the energy of which, arising from the high velocity of the extremely small but very numerous gas-molecules, is supplemented by their chemical action. The celebrated furnace Engineer FR. SIEMENS*) claims to promote radiation in his recent furnaces to such an extent that the gases, on leaving them, possess

*) Paper read at the London meeting of the iron and steel institute. October 1886.

only a small portion of heat which they afterwards give up by conduction to the heating surface of the boiler. Of late years his furnaces, chiefly applied to smelting purposes, in which radiation is the principle object, have all given excellent results, as well as the boiler furnaces designed by him on similar principles.

7) The determination of the height of a furnace is not amenable to calculation, for if we take it as large as possible for the sake of favourable combustion, we are met by the circumstance that the highly important action of radiation is inversely proportional to the square of the distance between the radiant body and the receiver of the heat radiated. So that in this case practical experience must be the guide, and has shewn that for Welsh, Westphalian, and similar coals, with natural draught, the furnace crown should be from 500 to 550 mm above the bars. Quite recently this dimension has been recommended by Mr. VOGT*), Chief Engineer of the Bergish Boiler Inspection Association, as based on the results of his exclusively practical observations extending over many years. For marine boilers, furnaces of 1.1 to 1.2 m in diameter have long been considered the best practice. A greater diameter than 1.3 m is stated to be useless for natural draught, though with forced draught under a heavy air pressure it may be of service, as the fires should, under these circumstances, be 30 cm or more in thickness. Recent practice (in cases where several boilers are required) has gone in the direction of fitting two furnaces of the usual diameter, rather than three narrower ones, as the reduced heating surface thus produced is more than balanced by the increased economy shewn by experience to arise from the more complete combustion of the coals. The fact that the old flat-sided boilers evaporated 8.5 to 8.7 ℔s of water per ℔ of coal, whereas the modern cylindrical high-pressure boiler can only evaporate 8.1 to 8.3 ℔s, is evidently only due to the roomy furnaces of the earlier boilers. Best height of furnace as found by experience.

8) A furnace of the above diameter and well adapted for short-flaming coals can only burn long-flaming ones with considerable losses of heat. Experience teaches that even in a roomy furnace, the sides and crown of which receive the impact of a voluminous flame, the internal part of the flame, not in contact with the furnace surfaces, never completely radiates its heat. The flame being perfectly permeable to heat, the walls Use of the same kind of fuel.

*) Zeitschrift des Vereines deutscher Ingenieure. 1887. P. 528.

of the furnace onght also to receive the full radiation of the internal parts of the flame. As this nevertheless is not the case in reality, it can only be explained, from what is said in 5), by the circumstance that the furnace surfaces are shrouded in a thick cloud of free carbon which keeps the heat-rays from them.

Object of lining the furnace.

9) β. **Lining the inside of the furnace with bad conductors of heat** has the advantage of preventing the withdrawal of heat from the flame, whereas the steel or iron furnace-plates of marine boilers, being good conductors, abstract a considerable portion of the heat from the gases of combustion. These plates, surrounded by water several hundred degrees cooler than the fire, must greatly reduce the temperature of the gases and thus set free a large amount of carbon (from the carbo-hydrides), which as already mentioned in 5) is lost for all calorific purposes.

Marine boiler furnaces with fire-clay lining.

10) Although the advantage of the furnace lining is well known (see also § 37,2), it has not as yet been much adopted for marine boilers; partly because of the increase of weight, partly on account of the risk of the lining collapsing in bad weather. Subsequently to the application of fire-clay lining to the flues of Cornish boilers, extending rather beyond the fire-space, as is done by FR. SIEMENS) with good results, similar linings have been fitted to marine boiler furnaces for burning liquid fuel. As the author has described in another place*) these linings have been used by WAGENKNECHT in Germany, AUDOUIN and d'ALLEST in France, SELWYN, SADLER, TARBUTT, and HENWOOD in England, and SPAKOWSKI in Russia. In the latter country URQUHART has had numerous locomotives at work since 1882, the fireboxes of which are completely lined, and the brickwork is reported to have stood well in spite of the shocks and oscillations to which it is exposed on the road. Besides these, VERDERBER in Hungary and von STORCKENFELD**) in Sweden have constructed locomotives having fireboxes lined with fire-brick.

Marine boilers with bricked-out furnaces.

11) Since 1879 a number of Belleville boilers (burning coals) with fire-brick lining to the furnaces, have been worked in France both in war-ships, as the Dispatch-boats "Voltigeur", "Hirondelle" and "Milan", as well as the Messageries Maritimes S. S. "Ortégal" and "Sindh", without any complaints arising as to accidents to the masonry at sea, &c. And in 1887 the trial trips of the U. S. Cruizer "Chicago" took place, the furnaces

*) Zeitschrift des Vereines deutscher Ingenieure. 1887. Plates XXXVII to XL.

**) Zeitschrift des Vereines deutscher Ingenieure. 1888. P. 1164.

(according to the drawings published*) being similarly bricked out and fired with coals, giving results in every way satisfactory.

12) These fire-clay or fire-brick linings serve as stores of heat and become white-hot on their inner surfaces, especially when liquid fuel is used, so that they not only do not reduce the temperature of the gases of combustion, but rather have the effect of igniting any free carbon passing over them, thus contributing largely to the perfect combustion observed with this system. Experience shews that if the diameter of the furnace is unduly reduced by the lining, the latter is rapidly worn out, because the flame, impinging hard upon it, has the destructive effect described in 6). In such cases the expense of renewing the lining may outweigh the economy due to the improved combustion of the fuel. **Destruction of the lining.**

13) γ. **The regulation of the air-supply** is supplementary to the enlarging and lining of the furnace. If the formation of smoke is to be prevented altogether, there must be a constant current of air through the grate of sufficient volume to completely burn the increased quantity of carbo-hydrides evolved from every supply of fresh fuel. This of course means that when the fire is burnt through and only the non-volatile portion of the coal (or coke) is being consumed, the air supply is excessive, and the loss of heat thus caused may under some circumstances be as great or greater than that due to the formation of smoke. (See further on this subject in 57.) For this reason, various plans have been devised for increasing the air supply after firing up and diminishing it when the fires have burnt through. Others again have been proposed for improving or intensifying the combustion. Among these devices we may distinguish **Object, and methods of attaining it.**

1) Dampers or ash-pit doors,
2) Valves fitted under the bridges,
3) Sliding shutters in the fire doors or furnace fronts,
4) Heating the air supplied to the fires,
5) Induced and forced draught.

The constructive details of the above are referred to in the division on furnaces and their fittings, only their objects and effects will here be described seriatim.

14) *Dampers or ashpit doors* close the ashpit mouth and thus shut off air from the fire. They are often fitted so that they can be regulated, like a throttle valve, to admit more or less air as required. **Object of dampers.**

*) Mittheilungen aus dem Gebiete des Seewesens. Pola 1884. Plate IV.

Small value of doors under the bridges.

15) *Doors under the bridges* form the back of the ash-pit and are usually worked by means of a rod from the boiler front. They are supposed to afford a means of supplying to the gases in the combustion-chamber the balance of the air which is necessary to completely burn them. Frequently the space between the back of the bridge and the combustion-chamber back is covered by a perforated plate, the holes in which are intended to distribute the air evenly throughout the combustion-chamber, an arrangement which is of very little use, apart from the holes soon becoming obstructed with soot, &c. Under natural draught, the combustion-chamber temperature is only about 600^0 to 700^0 and the cold air thus introduced would probably lower this still more, so that the free carbon which has once left the combustion zone (of the flame) cannot ignite. At the utmost a few incandescent fragments of coal may be completely burnt by contact with the extra air at the back. Some experiments made on the question in 1887 in England by SPENCE*) seem to corroborate this view. When all openings under the bridge were closed, the boiler shewed an average efficiency of 0.65; when the area of the openings was $\frac{1}{497}$ of the grate surface, the efficiency was 0.657; when the proportion was $\frac{1}{152.4}$, the efficiency was 0.658, so that it was only increased 0.8 % by this admission of air. SPENCE, however, states that vertical jets of air introduced into the hot gases either just in front of, or over, the bridge, are much more advantageous, but attended with practical difficulties and complications. By this means he increased the efficiency of his experimental boiler from 0.71 to 0.732, i. e. 2.2 %. FOTHERGILL'S**) forced draught arrangement, which includes the introduction of part of the blast in fine jets from the back of the boiler into the hot gases in the combustion-chamber, which he makes extra large, is said to have given good results.

Utility of admitting air above the bars.

16) *Shutters in the fire-doors or furnace fronts*, admitting air above the fire are of much more use than doors under the bridges. The air introduced in front has more time and more room to get mixed with the gases than it has when entering at the back; besides which, it is heated before it meets the gases and does not lower their temperature, thus increasing the

*) Transactions of the north-east coast institution of engineers and shipbuilders. 1888. Table I. Trials 1 to 12.

**) The Engineer. 1888. I. P. 274.

chance of a complete combustion as described in § 20, 8. This method of introducing air has since 1885 been carried out with particular success for marine boilers by HOWDEN in England, and after him, SACHSENBERG in Germany. Both use heated blast. SPENCE also experimented with the admission of air through the furnace front and thus increased the efficiency of his boiler from 0.671 to 0.71, nearly 4 %, with natural draught.

17) *Arrangements for heating the air on its way to the furnace* have been applied to marine boilers by HOWDEN*) and SACHSENBERG**). The blast which is introduced both below the bars, and over them through the furnace front, is made to traverse a nest of tubes fitted in the smokebox. By this means HOWDEN claims to have heated the air up to 200° or 220°, but judging from the results of SACHSENBERG'S experiments which only shewed a mean temperature of 95° to 100° during a voyage, — HOWDEN'S statement appears to be a considerable exaggeration. WYLLIE***) also fitted the S. S. "Stella" in 1885 with tubes in the smokebox through which the air passed to the fires, both above and below the bars, under induced draught. In this case however the warming of the air cannot have been so effective as in the former arrangement. Another instance is given by HOADLEY****) in America, who worked a land-boiler under induced draught, the air being heated to 170° in a box of tubes as in the other plans. He asserts that this arrangement raised the efficiency of the boiler from 0.688 in one case to 0.782, and in another to 0.814, thus saving from 10 to 15 % of coal. It appears however that the lion's share of this economy was more likely to be due to the moderate artificial draught than to the warming of the air. SPENCE also extended his experiments in the direction of warm blast. He heated the air in a system of tubes by means of a separate fire of coke and improved the efficiency of his boiler from 0.757 with cold blast, to 0.784 with the blast warmed to 100°; in the most favourable case the gain was 4.1 %. From these small figures we may infer *that heating the blast can only be of any economical value when the heat of the uptake gases is used for the purpose, thus avoiding any extra expenditure of fuel, and further when the arrangement of tubes &c. is unattended by any considerable outlay and is at least capable*

Practical value of heating the air, as found by various trials.

*) Transactions of the Institution of Naval Architects. 1886. P. 182.

**) Zeitschrift des Vereines deutscher Ingenieure. 1888. P. 586.

***) Transactions of the Institution of Mechanical Engineers. 1886. P. 489.

****) J. C. HOADLEY. Warm-blast steam-boiler furnace. New-York 1886.

of heating the blast to 100° C. The cost of maintenance of the tubes, which HOWDEN places in the smokebox, traversed internally by the air to be heated up to at least 100°, and externally exposed to the hot uptake gases of about 400° to 500°, must however always be considerable, judging from our experience as to the rapid destruction of superheater tubes (see p. 63 under 21).

Artificial draught in general.

18) *Artificial draught, whether forced or induced*, enables the air supply to be reduced almost to the theoretical quantity, thus bringing about a more complete combustion, which is the more favourable the more completely the air supply can be regulated by means of the arrangements referred to in the preceding pages.

Consequences of irregular firing.

19) **II. Defective firing** is most conspicuous when fresh fuel is supplied to the fires, and shews itself by the presence of smoke of greater density and extending over a longer time than is properly due to the quality of the particular coals used. In almost all cases of this sort, the fires are not replenished until they are nearly burnt down. Large quantities of fuel must then be thrown upon the grate, from which such a volume of carburetted hydrogen gases is suddenly evolved that the air-supply does not suffice for their combustion, so that carbonic oxide, carbo-hydrides, and particularly free carbon in the form of soot, escape unburnt. On the other hand, heat is absorbed in warming up the coals and driving off the carbohydrides, which diminishes the temperature of the gases, producing soot and causing carbonic oxide and other combustible gases to pass off in the smoke. As the fire-door remains open longer than it should, a large volume of cold air passes through the furnace, and by cooling it down, aggravates the above described tendencies towards the formation of soot. Besides all this, the cold air has a very injurious effect upon the highly heated parts of the boiler, as further referred to later on under the subject of the working of marine boilers.

Limitation of the losses of heat when firing up.

20) The following means have been tried in order to combat the losses of heat always attending the act of firing up with solid fuel to a greater or less degree:

a) pushing back the fire,
b) firing at regular intervals,
c) mechanical stokers.

For marine boilers however, the best means of attaining this object has been shewn by experience to be

d) the proper training of the firemen.

21) a. **Pushing back the fire,** i. e. displacing the burnt-through coal (or coke) towards the bridge and throwing the fresh or green coal on the bare space thus left at the front of the grate, enables the formation of smoke to be avoided. The carburetted hydrogen gases, formed at the front and mixed with sufficient air, are then obliged to traverse the hot zone of combustion at the back, so that no uncombined carbon can be liberated. The increased time however, during which the fire-door has to be left open to perform the operation of pushing back the fire, allows so much cold air to pass through the furnace as to withdraw more heat from it than would be lost by the liberation of carbon occurring in the usual method of firing practiced by good stokers.

Advantages and disadvantages of pushing back the fires.

22) Some comparative experiments made many years age, and reported by KNAPP*), with Saxon coals in a land-boiler, proved that better evaporation is to be obtained from the same quantity of coals, when the fires are treated as usual and smoke issues from the chimney, than when the fires are pushed back. This is the more easily understood when it is considered that the operation of pushing back the fire causes the loss of many unburnt fragments of coal and coke through the bars.

Knapp's experiments.

23) b. **A regular supply of fuel** without opening the fire-door, thus avoiding the admission of cold air, can only be attained with solid fuel by the use of a semi-gas furnace like TENBRINK'S or DONNELEY'S. In both of these the furnace resembles a well or shaft formed of masonry or water-tubes, supplied with coal through a funnel at the top, the ash and clinker being from time to time removed at the bottom, as required for the regular sinking of the burning fuel. As the fuel burns downwards and the gases liberated in the upper layers must pass through the lower ones which are in an incandescent and smokeless condition, the combustion is very good. DONNELEY'S furnaces of which further mention will be made later on have been at work since 1887 in several river steamers of the German Elbe Navigation Co. "Kette", and according to some comparative experiments of LEWICKI'S, are reported to save 22.32 % of the fuel consumed in ordinary furnaces. Unfortunately furnaces of this description require too much space for marine boilers, and the masonry surrounding the furnace proper is too heavy. Nevertheless the firm of PAPE & HENNEBERG in Hamburg intend to construct a similar furnace for marine purposes of a lighter and compacter design, trials of which are in prospect.

Donneley's furnace.

*) KNAPP. Handbuch der Technologie. III Edit. 1865. Table 347.

Kinds.

24) c. **Mechanical stokers** may be classified as those which
α) supplement the labour of the fireman, or
β) entirely replace it.
Of late years both kinds have been constructed and tried in increasing numbers.

Strupler and Cario's arrangements.

25) *α. Among the devices of the former kind*, which are intended to be to a certain extent independent of the skill possessed by the fireman, STRUPLER'S*) and CARIO'S**) have given good results with land boilers. The former scatters the coals evenly and lightly over the whole grate, while the latter curtails the time during which the fire-door is opened, by putting the fuel on with a tray similar to those used for charging gas retorts. On account of their filling arrangements both are useless at sea when the ship is rolling.

Uselessness of mechanical stokers at sea.

26) *β. Stoking arrangements of an exclusively mechanical nature*, intended to replace firemen altogether and only requiring trimmers to charge them and remove the ash, have been tried in a few merchant steamers. Some of them are reported to have worked well in good weather, and HENDERSON'S***) arrangement (fitted to a land boiler set in masonry) is credited with an economy of 33 % over hand firing. However, with all of them, some more or less serious hitch has always arisen as soon as bad weather is encountered. To provide against this, special fire-doors are in some cases provided, enabling ordinary firing to be reverted to. BACH****), after a lengthy description and criticism of the mechanical stoking arrangements shewn at the International Exhibition of smoke-preventing appliances in London of 1881, sums up thus, "it is possible to reduce the smoking of chimneys further than is done at present; it does not however appear worth while to adopt any of the devices for this purpose offered here". — This opinion is still applicable to the marine boilers of to-day although many of the modern mechanical stokers (to be described later on) have been improved upon since this exhibition.

Appliances for burning liquid fuels.

27) The only mechanical firing appliances which have answered well at sea both in good and bad weather are those used for burning liquid fuels in the form of spray. For these merely a boy is required to regulate the cocks for the oil supply and steam jet. This is the only case in which skilled firemen are not wanted on board.

*) Zeitschrift des Vereines deutscher Ingenieure. 1889. P. 47.
**) Ibid. P. 48.
***) Engineering 1889. II. P. 668.
****) Zeitschrift des Vereines deutscher Ingenieure. 1882. P. 91.

28) d. **Thorough training of the stokers,** upon whose skill and attention the working of marine boilers still depends in a very great measure, cannot be too frequently and urgently recommended. The influence of the *good* fireman, unfortunately not even yet sufficiently recognized, — upon the economical working of the entire machinery, is most strikingly illustrated by stoking-matches, such as those instituted by WALTHER-MEUNIER*) at Mühlhausen in 1880 and WEINLIG**) at Magdeburg in 1886. **Stoking-matches.**

29) *In Walther-Meunier's experiments*, ten practised firemen were each successively employed for 3 working days of 11 hours at the same boiler. The third day was not however taken into account, because by that time most of the firemen shewed signs of fatigue. About three tons of inferior Saar coals were burnt daily and the best fireman evaporated 6.17 kg of water per kg of dry coals (i. e. deducting the moisture), the worst fireman only 5.64 kg. The best of these ten firemen, who were all familiar for years with the working of boilers, thus produced 9.0 % more steam from the same weight of coals than the worst, or for the same quantity of steam raised he saved this amount of coal, which means about 300 kg of coal daily, or nearly 90 tons per year of 300 working days. **Walther-Meunier's experiment.**

30) *In Weinlig's trials* likewise, 11 skilled firemen competed, who had worked boilers for several years. The men were informed beforehand as to the conditions of the trial and had an opportunity of inspecting the coals, the boiler, and the furnaces. Each by himself worked the boiler one day in turn, the coals and feedwater being weighed. They had to keep up 45 ℔s of steam with the ordinary revolutions of the engine. The table on the following page gives the record made by the different firemen. **Weinlig's experiment.**

31) This table shews that of the eleven skilled firemen, the best man evaporated 6.89 kg of water per kg of coal, the worst only 4 kg, thus producing the enormous difference of 44 % in economy of coal. WEINLIG aptly remarks "compared with such differences in the economical handling of a boiler by the firemen, what is the use of all progress in the construction of furnaces, to which so much technical and scientific skill has been applied for years? In the whole of my long practice I do not know of a single case in which any improvement in a furnace achieved a saving of 40 % in the consumption." If trials with firemen of *long experience*, incited by a prize to **Inferences from the trials.**

*) Bulletin de la société industrielle de Mulhouse. 1880. P. 289.

**) Zeitschrift des Vereines deutscher Ingenieure. 1886. P. 123.

exert their utmost skill can produce such results, what sort of revelations might be expected from similar competitions among the ordinary stokers of sea-going steamers? In most merchant steamers any ordinary workman or labourer makes a trip or two as a trimmer, acquires only the very slight skill required for putting on coal, and is then rated as a fireman. In Continental navies, limited as they are to the period of service under conscription for training the stoker-recruits, matters are almost worse, as most of these men only become efficient firemen by the end of their time.

Results of Weinlig's stoking-match.

Stoker's number	Water evaporated per kg of coal kg	Average temperature of feed-water	Average pressure of steam in Atmos.	Average flue temperature	Average proportion of air-supply to theoretical quantity	Revolutions of engine per minute	Water evaporated per □ m of grate per hour kg	Remarks.
1	2	3	4	5	6	7	8	9
1	6,89	22,0°	3,07	233°	3,1	69,8	7,2	1) The temperature of the gases was taken near the damper. 2) The gases, for calculating the figures in col. 6 from the smoke analysis, were drawn from the back end of the flue tube of the boiler. 3) The grate-surface was intentionally made too large for its work, in order to test more distinctly the skill of the firemen in arriving at the best thickness of fire to suit the draught.
2	6,81	23,5	3,10	230	3,0	80,0	7,8	
3	6,64	40,0	3,20	217	3,2	69,9	7,7	
4	6,43	37,0	3,09	250	3,4	78,0	8,1	
5	6,01	33,0	3,00	198	2,3	78,0	6,4	
6	5,64	29,5	3,15	250	3,8	74,0	7,5	
7	5,49	29,5	2,80	240	4,1	73,0	7,5	
8	5,40	23,0	3,50	264	3,3	78,0	8,4	
9	5,00	36,0	2,93	255	3,8	68,0	8,4	
10	4,80	24,5	3,20	252	3,2	75,0	6,9	
11	4,00	27,0	3,16	298	5,1	84,0	7,7	

Stokers in the U. S. Navy.

32) How strong the tendency formerly was to accept any man for a stoker who was strong enough to throw a shovelful of coals on the fire, is shewn by an order made several years ago by the U. S. government, quoted by SENNETT,*) to the effect that in war-ships seamen should be told off for duty in the stokehole. After the actual carrying out of this measure, the Bureau of Engineering of the U. S. Admiralty resolved in 1873 that, on account of the annoyances, particularly the wasteful and disappointing results, consequent upon the above order, in future only special stokers, exclusively for duty below, should be shipped; effect has since been given to this resolution.

Seamen-stokers in the French Navy.

33) The French naval authorities**), compelled by necessity, have recently had to make up their minds to train seamen as stokers,

*) R. SENNET. The marine steam engine. London 1885. II edit. P. 57.

**) Mittheilungen aus dem Gebiete des Seewesens. Pola 1889. P. 78, from the Bulletin officiel de la Marine.

because the increased numbers of suitable men required could no longer be supplied by the ordinary process of recruiting, as may be pretty clearly inferred from the ministerial order. Vigorous seamen, not less than 18 years of age and 1.58 m high, have to undergo a course of instruction in firemen's duties on board a training-ship, and after sufficient practice in tugs and harbour steamers, are drafted to ships in commission. Having kept 60 watches on the fires, the best of the men are promoted to "uncertificated seamen-stokers". Only those who, after a year's service in the boiler-room have shewn themselves to be at least *good* firemen, are eligible for undergoing their examination as stokers. Having succeeded in this, they become "certificated seamen-stokers", and draw the same pay as actual stokers, whom they are intended in every way to replace. The certificate of a "torpedo-boat seaman-stoker" is not much more difficult to obtain. After what has been said there is little doubt as to the sort of work likely to be got out of such stokers, who have never learnt any trade as smiths, fitters, &c., besides having been chosen from the most ignorant rank of seamen, unable either to read or write. They will hardly contribute to increased efficiency of the boilers. In the British Navy a very sensible want of practised stokers is reported to have shewn itself at the celebrated review before the German Emperor and during the manœuvres which followed.

34) When we consider that the extended trials of war-ships, particularly when under forced-draught, have long ceased to be a test of the capacity of their machinery and are becoming more and more a trial of or rather a tremendous strain upon the physical powers of the engine-room staff, especially the stokers, the question is certainly not out of place: "*why should we be perpetually struggling after higher pressures, artificial draught, improved engines and valve-gears, &c., while the principal factor in getting the highest possible, as well as the most economical, power out of the machinery remains so astonishingly untrustworthy?*" Value of stokers.

35) The only hope of improving this unhappy state of affairs is in the formation of special schools for stokers, in which the boys not only acquire the necessary manual skill by practical training before the boilers, but are also taught by a methodical course of instruction to properly comprehend the stoker's art. Stokers' schools.

36) *The manual skill of the stoker* comprises handiness, particularly rapidity, in opening the fire door, firing up, slicing, pricking, Manual skill of the stoker.

cleaning fires, getting up ashes, sweeping tubes, working the damper, &c.

The stoker's art.

37) *The stoker's art*, on the other hand, consists principally in the observance of the following rules:

1) In the furnace, that temperature is to be maintained which is necessary for the ignition of the fuel and the combustion of the gases disengaged from it.
2) In order to keep up this temperature, a certain quantity of heat must remain stored up either in the layer of fuel itself, or in the fire-proof lining of the furnace.
3) The thickness of the fire must be increased in proportion to the weight of fuel to be burnt, i. e. to the weight of steam to be produced in unit time.
4) The velocity with which the air for combustion is conducted to the fire, — *the draught*, — must be in proportion to the thickness of the fire.
5) In an ordinary bar-grate the thickness of the fire is constantly varying as the fuel burns down and is renewed, so that the draught requires perpetual regulation; and this must be very carefully attended to, as a portion of the air spaces between the bars gradually clinkers up, thus diminishing the air-supply.

So long as the direct application of solid fuels is retained, and until the whole arrangement of the marine boiler furnace is altered into something of the nature of a gas-furnace, the personal capability of the fireman will continue to play a more and more prominent part.

Sources of these losses.

38) **III. Losses of heat from defective combustion of the fuel** are brought about

a) by fragments of fuel falling into the ash-pit,
b) by the formation of soot,
c) by gases passing off unburnt.

Average magnitude of these losses.

39) a. **The losses from unburnt fragments** are in general not very important. They arise either from "small" falling through the bars immediately on firing up, or from splinters flying off from the burning coal and becoming lost in the same way, or again by small pieces of partially consumed coal either escaping through the bars during slicing, or becoming imbedded in clinker. The particles of fuel thus wasted do not however amount to more than from 5 to 8 % of the total consumption except under unfavourable conditions, such as with coal which is brittle and full of small, or where the fires are constantly disturbed by unskilful firing. For war-ships at cruising speed it is advisable

in the interest of increased economy to pick out the useful fragments of cinder from the ashes (see § 20, 54) as they are easily distinguished, and put them on the fires again. According to SENNETT *), experiment has shewn that with the above process and careful handling of the fires, the unconsumed particles of fuel may be brought down almost to nothing, or at any rate within $2^1/_2$ % of the total weight burnt.

40) In exact scientific experiments, the loss due to unburnt fragments of coal is computed from an average sample of the contents of the ashpit, obtained by a similar process to that described on p. 174 for testing coals. The sample (say 5 grammes) is well pulverized, accurately weighed, and strongly calcined for a considerable time. The loss of weight subsequently ascertained shews of course the proportion of unburnt carbon contained in the ashpit residue. If the latter amount to $2^1/_2$ % of the coal burnt and if the loss of heat on calcining the 5 gramme sample is 1 gramme or 20 %, then there are 0.5 kg of unconsumed carbon in the ash from 100 kg of coal. If this 0.5 kg were burnt to carbonic acid it would produce by Eq. 92^a or 92^b, p. 175, **Exact determination of the losses of heat from unburnt fragments of fuel.**

$$8000 \times 0.005 = 40 \text{ T. U. or } 8100 \times 0.005 = 40.5 \text{ T. U.}$$

With very careful stoking, the loss from unconsumed fragments may become so small as to be negligible, as these figures shew.

41) b. **The losses of heat due to the formation of soot** are likewise very small with good firing and scarcely ever exceed 1 % of the absolute calorific power. MINARY**) suggests that the amount of these losses may be accurately determined by drawing off a sample of the uptake gases through a combustion tube provided with a layer of fragments of asbestos about 10 cm thick, burning the soot deposited from the gases in a current of oxygen, and computing the weight of the soot from the quantity of the carbonic acid thus produced. If, for instance, 1 cbm of the gases contains 0.0011 kg of soot and if, as calculated in 42), 1 kg of coal gives 7.294 cbm of dry gases, the loss in soot per kg of coal is $7.294 \times 0.0011 = 0.008$ kg of carbon, equivalent to **Magnitude and calculation of these losses.**

$$8000 \times 0.008 = 64 \text{ T. U.}$$

or not quite 0.88 % of the absolute calorific power of the coal, supposing that all the soot could have been burnt to carbonic acid. As only a comparatively bad furnace would produce the amount of soot assumed above and the loss of heat due

*) R. SENNETT. The marine steam engine. London 1885. II edit. P. 53.

**) F. FISCHER. Taschenbuch für Feuerungstechniker. Stuttgart 1883. P. 24.

to soot may be as small as 0.1 % of the absolute calorific value in a good furnace, this loss is usually disregarded in practice.

Estimation of the dry gases.

42) c. **The losses of heat due to gases escaping unburnt,** which with defective combustion, may represent 20 to 30 % of the absolute calorific power, shrink to 1 % in good furnaces, as the table on p. 206 shews. F. FISCHER*) gives the following example which clearly brings out the importance of these losses. Suppose that a bad furnace produces from average coals (§ 19, 13) ash as described in 40) and soot as in 41), as well as shewing the following smoke-analysis,

Carbonic acid	15 %	together 20 %.
" oxide	4 "	
Marsh-gas	1 "	
Hydrogen	1 "	
Oxygen	2 "	
Nitrogen	77 "	

then of the 0.8 kg of carbon contained in every kg of these coals, 0.005 kg remain in the ashes, so that only 0.795 kg of carbon $= C$ are burnt. These produce by formula 103^b, p. 192,

$$V_r' = \frac{0.795}{0.54 \times 0.2 + 0.0011} = 7.294 \text{ cbm}$$

of dry gases; because 20 % of the gases contain carbon and therefore $k = 0.2$, and further because in every cbm of these gases there are not only 0.54 kg of carbon held in combination, but also 0.0011 kg mechanically suspended as soot.

Calorific value of the unburnt gases.

43) The 7.294 cbm of dry gases consist of

$$\text{Carbonic acid} = 15\,\% = \frac{7.294 \times 15}{100} = 1.094 \text{ cbm}$$

$$\text{" oxide} = 4\,\% = \frac{7.294 \times 4}{100} = 0.292 \text{ "}$$

$$\text{Marsh-gas} = 1\,\% = \frac{7.294 \times 1}{100} = 0.073 \text{ "}$$

$$\text{Hydrogen} = 1\,\% = \frac{7.294 \times 1}{100} = 0.073 \text{ "}$$

$$\text{Oxygen} = 2\,\% = \frac{7.294 \times 2}{100} = 0.146 \text{ "}$$

$$\text{Nitrogen} = 77\,\% = \frac{7.294 \times 77}{100} = 5.616 \text{ "}$$

The 2 % of oxygen is a sign of too great excess of air, carbonic acid and nitrogen are non-combustible, so that of combustible gases there escape

*) F. FISCHER. Taschenbuch für Feuerungstechniker. Stuttgart 1883. P. 28.

Carbonic oxide	$= 0.292$ cbm $= 0.292 \times 1.2593 = 0.368$ kg,	which burnt to	CO_2	by § 20, 11, produce	$0.368 \times 2400 = 893$	T. U.
Marsh-gas	$= 0.073$ cbm $= 0.073 \times 0.7160 = 0.052$ kg,	„ „	$CO_2 + 2H_2O$	„ „	$0.052 \times 11900 = 619$	„ „
Hydrogen	$= 0.073$ cbm $= 0.073 \times 0.0896 = 0.0065$ kg,	„ „	H_2O	„ „	$0.0065 \times 2900 = 188$	„ „

The losses therefore together amount to *1700* T.U.

or $\frac{1700 \times 100}{7300} = 23\,\%$ of the absolute calorific power of the coals. With good combustion the uptake gases of these average coals would contain only traces of marsh-gas and hydrogen and perhaps about $0.1\,\% = 0.0073 \times 1.2593 = 0.0092$ kg of carbonic oxide, which carry off $0.0092 \times 2400 = 22$ T. U. or only 0.3 % of the absolute calorific power of the coals.

44) **The total loss of heat due to unburnt components of the fuel** in the above example is made up as follows **Total loss of heat due to unburnt components.**

Ash	40	T. U.
Soot	64	„
Unburnt gases	1700	„
Together	1804	T. U.

or nearly 25 % of the absolute calorific power of the coals in this bad furnace, whereas in a good furnace only about 0.55 % of the absolute calorific power is lost from ash, 0.1 % from soot, and about 0.3 % from unburnt gases, together only about 1 % from all unconsumed components of the fuel.

45) **IV. Losses of heat due to the high temperature of the residue of combustion** are caused **Losses of heat from residue.**

a) by the ash and clinker which fall into the ashpit,

b) by the escaping funnel gases.

46) a. **The losses of heat from the high temperature of the ashes** are as a rule so small that they only require attention in accurate scientific experiments. Even if the ashes are 800° hotter than the external air, which is not usually the case, as they lie immediately upon the bars, — and assuming 10 % of ash possessing a specific heat of 0.25, we get per kg of coal only $0.1 \times 800 \times 0.25 = 20$ T. U., or not quite 0.3 % of the absolute calorific power of the coal. But even this amount of loss is scarcely incurred in reality, as most of the heat in the ashes is given up to the air on its way through the ashpit to the bars. **Heat contained in the ashes.**

47) b. **The losses of heat due to the high temperature of the gases leaving the funnel** are unavoidable because a "natural draught" depends chiefly upon the difference between the weight of the hot gases in the funnel and that of a column (of the same dimensions as the funnel) of the external air. This difference of weight again depends upon the difference of temperature of the two columns. The quantity of heat lost or carried off in the funnel gases is, however, usually greater than is necessary to **Keeping up the funnel draught.**

sustain the draught. The most effective action of the funnel is reached when the greatest possible *weight* of gases is discharged by it in unit time. According to HERRMANN*), this is found to take place when the absolute temperature T_1 of the gases at the *mouth* of the funnel is twice that of the external air T, whence follows, assuming the usual boiler room temperature of $t = 20^0$,

$$T_1 = 2\,T = 2\,(273 + 20) = 586^0$$

or

$$t_1 = 586 - 273 = 313^0.$$

According to RANKINE**)

$$T_1 = 2^1/_{12}\,T = 2^1/_{12}\,(273 + 20) = 610^0$$

or

$$t_1 = 610 - 273 = 337^0.$$

In practice the mean of these two temperatures, $\frac{313 + 337}{2} = 320^0$, is generally taken as the most effective temperature for the funnel gases, so that they are heated $t_1 - t = 320 - 20 = 300^0$. If the temperature of the gases is raised above this limit, their velocity of efflux is increased, it is true, but at the same time their volume, and at a higher ratio; so that the *weight* of gases discharged in unit time, or in other words, the draught, is diminished. With artificial draught, however, the gases may leave the funnel at a lower temperature, dependent upon that of the boiler steam.

Combination of the losses of heat.

48) The losses of heat Q due to the rise of temperature of the funnel gases above that of the external air may be divided into

α) those due to the aqueous vapour, Q_1

β) „ „ „ „ dry gases, Q_2.

Heat of the aqueous vapour with complete combustion.

49) *α. The heat Q_1 lost in the aqueous vapour per kg of fuel* is calculated from the weight W_r of the vapour as found by Eq. 101 or 101ᵃ on p. 189, or from its volume W_r', by multiplying either of these quantities with the specific heat of steam, $c = 0.4805$ for weight, or $c' = 0.3870$ for volume, respectively, and also by the rise of temperature $t_1 - t$, thus:

$$Q_1 = c\ W_r\,(t_1 - t) \text{ T.U. per kg of vapour} \quad (104)$$

$$Q_1 = c'\ W_r'\,(t_1 - t) \text{ T.U. „ cbm „ „} \quad (104^a)$$

For the numerical values in § 20, 43, viz $W_r = 0.568$ kg and $W_r' = 0.705$ cbm per kg of average coals, we therefore get

$$Q_1 = 0.4805 \times 0.568 \times 300 = 82 \text{ T.U.}$$

$$Q_1 = 0.3870 \times 0.705 \times 300 = 82 \text{ „}$$

Heat of the aqueous vapour with defective combustion.

50) It is assumed, in the above estimations of W_r and W_r', that all of the hydrogen in the coals is burnt to H_2O. With

*) J. WEISSBACH'S Ingenieur- und Maschinen-Mechanik, bearbeitet von G. Herrmann. Braunschweig 1883—87. Edit. V. Vol. II. P. 882.

**) J. W. M. RANKINE. A manual of the steam engine. London 1873. P. 289.

defective combustion as referred to in 42) on p. 212, this is not the case, but on the contrary, hydrogen and marsh-gas escape unburnt, thus reducing the quantity of aqueous vapour. By 43) the funnel gases contained 0.073 cbm of free hydrogen and 0.073 cbm of marsh-gas which latter always holds in combination 2 cbm of hydrogen per cbm, giving $2 \times 0.073 = 0.146$ cbm of hydrogen; so that altogether $0.073 + 0.146 = 0.219$ cbm, or $0.219 \times 0.0896 = 0.02$ kg, or 2 % i. e. half of the 4 % of hydrogen in these average coals, is lost in an unburnt state. These 0.02 kg of hydrogen would have produced $9 \times 0.02 = 0.18$ kg or $\frac{0.18}{0.8084} = 0.22$ cbm of aqueous vapour, so that *ceteris paribus* there are no longer $W_r = 0.568$ kg or $W'_r = 0.705$ cbm, but only $0.568 - 0.18 = 0.388$ kg or $0.702 - 0.22 = 0.482$ cbm of aqueous vapour present in the funnel gases. The losses of heat from aqueous vapour are therefore in this case

$$0.4805 \times 0.388 \times 300 = 56 \text{ T. U.}$$
$$0.3870 \times 0.482 \times 300 = 56 \text{ „}$$

51) *β. The heat Q_2, lost in the dry gases,* is found by multiplying the quantity v of each of the different gases by its specific heat c_0 and by the rise of temperature $(t_1 - t)$ and afterwards adding these products **Heat of the dry gases with a low excess of air-supply.**

$$Q_2 = \Sigma\,[c_0 v (t_1 - t)] \text{ T. U.} \quad \ldots\ldots\ldots\ (105)$$

The following table shews the computation of the heat losses per kg of coal, assuming the analysis of the funnel gases to correspond with 42), and therefore their volumes with 43) and the rise of temperature to be 300°.

Gas	Volume v of the gas in cbm	Specific heat, c_0 of the gas in T. U.	Rise of temperature of the gas in °C.	Loss of heat $300\ c_0\ v$ in T. U.
1	2	3	4	5
Carbonic acid .	1,094	0,463	300	152
„ oxide .	0,292	0,308	300	27
Marsh-gas	0,073	0,424	300	9
Hydrogen	0,073	0,305	300	7
Oxygen	0,146	0,311	300	14
Nitrogen	5,616	0,306	300	517

So that the loss of heat due to dry gases, $Q_2 = 726$ T. U.

Under the above defective conditions of combustion, in which the excess of air is very small, as shewn by the 2 % of free oxygen, the loss of heat in the dry gases is nearly 10 % of the absolute calorific power.

Heat of the dry gases with a greater excess of air.

52) This particular loss of heat increases in direct proportion to the excess of air-supply, as may be seen by substituting in the calculation the quantities of the different gases in § 20, 45, p. 190. These gases were produced by the combustion of 1 kg of average coals under an air-supply 1.3 times as great as that theoretically necessary. There was 5 % of free oxygen and the result of the smoke analysis is

Gas	Volume v of the gas in cbm	Specific heat, c_0 of the gas in T. U.	Rise of temperature of the gas in °C.	Loss of heat $300\, c_0\, v$ in T. U.
1	2	3	4	5
Carbonic acid .	1,483	0,463	300	205
Oxygen	0,494	0,311	300	46
Nitrogen	7,909	0,306	300	726
Sulphurous acid	0,014	0,445	300	2
			Together	$Q_2 = 979$ T. U.

or about 13 % of the absolute calorific power. The loss due to sulphurous acid is so small that it can be neglected, especially as it is usually determined as if it were included in the carbonic acid.

Total loss of heat Q from the rise of temperature of the gases.

53) **The total loss of heat Q, due to the rise of temperature of the funnel gases,** is therefore, by the preceding

$$Q = Q_1 + Q_2$$

$$Q = [c\, W_r + \Sigma\, (c_0\, v)]\, (t_1 - t) \text{ T. U.} \quad \dots\dots\dots \quad (106)$$

which gives, for the examples chosen

with low excess of air supply $726 + 56 = 782$ T. U.

„ greater „ „ „ $979 + 82 = 1061$ „

i. e. in the former case a loss of nearly 11 % and in the latter about 14.5 % of the absolute calorific power of the average coals.

Approximate estimation of the loss Q.

54) *The loss of heat Q may be approximately determined* from Eq. 106^a, below, if we substitute in Eq. 106 as the weight of aqueous vapour

$$W_r = W + 9\, H,$$

neglecting the last term of Eq. 101, p. 189, and put $c = 0.48$, so that

$$c\, W_r = 0.48\, (W + 9\, H);$$

further — estimating the volume of the dry gases by the approximate formula 103^b, p. 192 as

$$V'_r = \frac{100\, C}{0.54\, k},$$

and their mean specific heat at 0.32 per cbm, we get

$$Q = \left[0.32\, \frac{100\, C}{0.54\, k} + 0.48\, (W + 9\, H)\right] (t_1 - t) \text{ T. U.} \quad \dots \quad (106^a)$$

In this formula, which gives the heat lost in the chimney gases as determined in steam boiler trials conducted according to the rules of the Verein Deutscher Ingenieure,

C is the kg of carbon per kg of fuel,
H „ „ „ „ hydrogen „ „ „ „ ,
W „ „ „ „ hygroscopic water „ „ „ „ ,
k „ „ percentage-volume of carbonic acid in the chimney gases.

55) Applying this formula to gases of the composition given in 52), we find the loss is Example.

$$Q = \left[0.32\,\frac{100 \times 0.80}{0.54 \times 15} + 0.48\,(0.03 + 9 \times 0.04)\right] 300 \text{ T. U.}$$

$$Q = (3.184 + 0.1872)\,300 = 1011 \text{ T. U.}$$

that is, 51 T. U. less than the exact value determined in 53), which is due to neglecting the moisture of the air and the free oxygen in the gases in the approximate formula 106ª.

56) *The loss of heat Q may be shortly arrived at* by assuming that 1 kg of fuel produces, by complete combustion, 1 kg of chimney gases, to which v times the theoretical weight of air is added. The mean specific heat of the $vL + 1$ kg of gases is taken in practice at 0.25. So that for a rise of temperature of $t_1 - t$, the loss becomes Short method of calculating Q.

$$Q = 0.25\,(vL + 1)\,(t_1 - t) \text{ T. U.} \quad \ldots\ldots\ldots \quad (106^b)$$

For 1 kg of average coals burnt with 1.3 times the theoretical weight of air L, that is 10.42 kg as calculated in § 20, 26, p. 180, and for $t_1 - t = 300°$,

$$Q = 0.25\,(1.3 \times 10.42 + 1)\,300 = 1090 \text{ T. U.}$$

or only 29 T. U. greater than in 53), therefore this formula is near enough for most cases. Just as the approximate formula 106ª gives Q too small, the simple formula 106ᵇ always makes it appear too great, in consequence of the assumption that the combustion is complete and unaccompanied by ash.

57) From the last formula (106ᵇ), it is evident that the loss due to the rise of temperature of the gases increases in direct proportion to v, which denotes the air supply, whereas on the other hand, the loss due to the escaping of gases unburnt increases as the air supply is diminished. Some very instructive experiments were made by BUNTE*) at Munich with the object of ascertaining the exact amount of air supply which would bring the sum of these two losses to a minimum. These experiments are illustrated by Figs. 1 and 2, Plate 5. The most favourable air-supply.

*) Wochenschrift des Vereines deutscher Ingenieure. 1880. P. 169.

They were carried out with exactly the same kind of Ruhr coal, burnt in the same furnace with a varying air-supply on an ordinary grate. The vertical lines, representing the different experiments, are arranged in succession in such a manner that the greatest air-supply (about twice the theoretical), and consequently the lowest proportion of carbonic acid (8 %) in the chimney gases, corresponds to the first experiment (1), while at the last experiment (9), the coal was burnt with about the theoretical air-supply and the proportion of carbonic acid in the gases was 16.5 %. The whole length of the ordinates shews the total calorific value of the fuel, a varying portion of which, distinguished by shading, was lost under the particular conditions of each experiment. The fraction (of the whole heat) which was usefully turned to account in the boiler is represented by the remaining portion of the ordinates. The loss due to ash is neglected.

Inferences from Bunte's experiments.

58) These experiments of BUNTE'S shew clearly that with a large air-supply the loss due to unburnt gases is very small (about 1 to 2 %), but that the heat carried off by the gases and lost up the funnel is then proportionately great. This latter loss is reduced in proportion as the air-supply is diminished (as shewn in Fig. 2) but then, under similar circumstances, the loss due to unburnt gases is increased, and reaches 8, 10, and even 15 %; particularly when, as in experiments (7) and (9), the fire is kept about 20 cm (or 8″) thick. *For practical conditions, i. e. to use the calorific power of the fuel to the fullest extent, that combustion is the most favourable under which the sum of the losses of heat due (1) to unburnt gases and (2) to the rise of temperature of the gases, is a minimum. Ceteris paribus, the smaller the excess of air with which this favourable combustion can be brought about, the greater will be the useful calorific power of the fuel.*

Combustion with and without smoke.

59) The particular quantities of air-supply which, by the preceding, are the most favourable for different kinds of coal are given in § 20, 27. In their application, unconsumed gases will of course escape from the chimney, — in other words, there will be smoke. The heat lost in the smoke, together with the heat carried off in the hot gases will not cause so great a loss as that occurring with a large air supply and without smoke, as the following example teaches. In a fire supplied with only the theoretical quantity of air, the loss due to unburnt gases may be estimated, according to BUNTE'S experiments, at not more than 10 % of the absolute calorific power of the fuel. For average coals this means a loss of

730 T. U., to which must be added the loss from the rise of temperature, amounting by Eq. 106[b] to $0.25\,(10.5 + 1)\,300 = 861$ T. U., together 1591 T. U. Even when the air-supply is equal to twice the theoretical quantity, about 1.5 % of the gases escape unburnt, representing 109 T. U., while the loss due to rise of temperature is $0.25\,(2 \times 10.5 + 1)\,300 = 1650$, or together 1759 T. U., against 1591 T. U. — the loss with very imperfect combustion. The difference is 168 T. U. or 2.3 % of the absolute calorific power in favour of the latter. *A smoking fire may therefore be more efficient than a smokeless one, and it may well be said that combustion devoid of smoke and soot is to be commended from a hygienic point of view, but not always from an economical one.*

60) **V. The losses of heat due to radiation and convection** will always be rather greater in marine boilers, even when cleaded, than in land boilers, well protected by masonry. For the latter they are, according to F. FISCHER *), about 8 to 10 % of the absolute calorific power, and although they are more considerable in marine boilers, they may be to a great extent overcome by good cleading, the use of baffle plates on fire doors and furnace fronts, air casings on smokebox doors, and similar arrangements for reducing radiation. These losses were formerly looked upon as much more serious than they are in reality, — according to recent experiments. The method adopted has been to determine first the absolute calorific power of the coals with a calorimeter, next, the useful calorific power from the water evaporated, and afterwards to calculate the losses from an analysis of the ash and funnel gases as just described. The difference between the absolute calorific power on the one hand, and the useful calorific power plus the various losses as above on the other hand, shewed the loss due to radiation and convection. The amount of this loss is given in the following table, the experiments referred to being all made with coals, except those in Hanover, where charcoal was used. KENNEDY'S experiment was carried out with one of THORNYCROFT'S recent water-tube boilers as fitted in his torpedo-boats. It must be said that the extremely small loss from radiation and convection excites a certain degree of doubt as to the efficiency of this boiler, which is given as nearly 87 %.

Magnitude and method of estimating the losses of heat due to radiation and convection.

61) It is easily seen from the preceding that even in marine boilers of the best design and carefully fired, the losses of heat under

Average values for the useful calorific power.

*) Zeitschrift des Vereines deutscher Ingenieure. 1886. P. 47.

Table on evaporative experiments.

Place	Mühlhausen		Essen		Hanover		London
Experimenter	Scheurer-Kestner		F. Fischer, Böcking and Vogt		F. Fischer and Weinlig		Kennedy*)
Date	1869	1885	1883	1883	1885	1885	1889
1	2	3	4	5	6	7	8
Coal burnt per □ m of grate in kg	—	—	123,8	86,3	46,9	55,4	—
Water evaporated per □ m of heating surface . . . . in kg	—	—	26,6	24,7	8,7	8,9	—
Absolute calorific power of the coals, determined calorimetrically, T. U.	—	—	7790	7720	7630	7180	—
Heat imparted to the boiler water %	60,5	67,3	74,9	68,4	83,6	76,0	86,8
Loss in the ash . . . . %	1,5	—	2,1	3,6	0,9	0	—
Loss due to imperfectly burnt gases %	14,5	3,1	—	—	0,3	4,9	0,5
Loss due to the rise of temperature of the funnel gases . . . . . %		8,5	16,7	19,3	10,6	10,9	10,8
Loss due to radiation and convection, remainder . . . . %	24,5	21,1	6,3	8,7	4,6	8,2	1,9

the most favourable conditions amount to from 15 to 20 % of the absolute calorific power and that with bad boilers and unskilful stoking they may rise to 50 %. It thus appears that for the kinds of coal ordinarily used at sea, the average absolute calorific power of which is about 7500 T. U., we can only estimate the useful calorific power in the most favourable cases at 6000 T. U., and under the worst conditions at about 4000. As a generally applicable average value we may take 5000 T. U.

Concluding observations.

62) The improvement of the furnace, in the widest sense of the word, and the augmentation of the evaporative power of the boiler, which can only be attained by a thorough comprehension of the conditions influencing complete combustion, as well as of the causes of heat losses, form one of the chief aims of modern marine engineering. The important achievements of never-resting inventiveness will, for the future, only be such as tend towards increasing the economical efficiency of the boiler — not, as many think, towards raising the steam pressure.

§ 22.

Characteristics of good marine coal.

General qualities.

1) **I. Desirable qualities of coals.** Of a good coal, particularly for naval purposes, a number of qualities are demanded, which no single kind of coal completely possesses. Such coals should

*) The Engineer. 1889. II. P. 437.

therefore be selected as are distinguished by as many as possible, or at any rate by the most important, of the following characteristics.

a) high calorific power,
b) high density,
c) high tenacity,
d) low percentage of ash,
e) little smoke,
f) easy inflammability,
g) slight tendency to cake.

2) a. The coals should possess a **high calorific power** in order that as much water as possible may be evaporated with a given quantity of coal, and the vessel propelled the longest possible distance by the contents of her bunkers. The useful calorific power, which depends both upon the absolute calorific power of the coals and the efficiency of the furnace, comes out in average working at sea at about Calorific power.

6 kg of water per kg of coal,	with inferior coal and dirty boilers,
7 " " " " " " "	with average coals and not quite clean boilers,
8 " " " " " " "	with good coals and clean boilers,
9 " " " " " " "	with the best coals and new boilers,
10 " " " " " " "	with the best coals, new boilers, and feed-heaters.

A ratio of evaporation of 9 or 10 to 1 is only attainable on trial trips under very favourable conditions. According to the yearly abstracts in my possession of 28 mail steamers, the best result obtained at sea with the old flat-sided boilers was an average of from 7.5 to 7.8 to 1; declining to the worst at 5.5 to 1 in a cylindrical boiler with narrow furnaces. In the evaporative experiments at Wilhelmshaven*) a ratio of from 8.3 to 9.3 was reached with such descriptions of Westphalian coal as were found suitable for naval purposes, while only an average of 8.5 to 1 could be got out of the Welsh coal tried at the same place.

3) b. **Great weight** is a necessary quality, so that the greatest possible quantity may be stowed in the available space on board. At the Wilhelmshaven experiments the weight of the coals Weight.

*) Beilage zum Marine-Verordnungsblatt Nr. 23 for 1886.

was determined by breaking them into pieces weighing not over 0.5 kg which were afterwards sifted, to separate the small, in a sieve having meshes of 30 mm and inclined at 40°; the pieces were then packed in a box of exactly 0.25 cbm contents. Four such boxes of course gave the weight per cbm of the pieces of coal. This weight, for the coals most suitable for marine purposes, came out at from 730 to 800 kg, so that 1000 kg would take up a space of from 1.25 to 1.37 cbm. But with careful trimming in roomy bunkers, one metrical ton (1000 kg) of coals of average density, fresh from the pit, can be stowed in 1.2 cbm. (See also § 19, 13.)

Tenacity.

4) c. **High tenacity** is also a necessary quality, in order that too much small may not be formed during loading, transit, shipping, and getting out of the bunkers. — At Wilhelmshaven the tenacity of the coals was tested by putting 50 kg at a time in an iron drum 0.85 m in diameter, having three radial plates 16 cm broad attached to its internal periphery. The drum was caused to revolve steadily fifty times in two minutes, and its contents were sifted (see 3). The weight of the pieces which did not pass through the sieve, expressed as a percentage of the 50 kg, was taken as a measure of the tenacity. This number was found to vary from 30 to 90 for the different kinds of coals. The best marine coals only have a tenacity number between 40 and 55. Westphalian coals, possessing the highest calorific power, have like good Welsh coals, a tenacity number between 44 and 48. It is worthy of note that Westphalian bricquettes, or "patent fuel", have invariably a higher tenacity than natural coal of equal calorific power; the tenacity number of patent fuel varies mostly between 60 and 70.

Residue.

5) d. **Little residue** should be produced, because the constituents of it uselessly take up space in the bunkers, thus limiting the room for the actual fuel. Besides, coals forming much clinker are difficult to work on the grate and those with a high percentage of ash are troublesome on account of the large quantities to be got overboard. — At Wilhelmshaven the unburnt residue on the bars was collected during the day following each trial, weighed, and taken into account as ash. The contents of the ash-pits, and the soot and cinders in the smokeboxes and tubes were likewise collected, weighed, and expressed as a percentage of the total coal burnt. The minute quantity of fragments of unburnt coal in the clinker and ash was neglected. The unburnt residue, determined in this manner, varies for the different kinds of coals, between 3 and 20 %, but only

those, the residue of which does not exceed 8 %, can be called suitable for marine boilers. For most of the Westphalian patent fuel which was found suitable, the proportion of residue was between 6 and 8 %, while some kinds shewed less than 6 %.

6) e. **Slight smoke, and this of as light a colour as possible,** should be produced, in order to avoid soiling the ship and inconveniencing those on deck, especially the commander. In war it is besides particularly important that steam ships should burn coals as smokeless as possible, so that the vessels may remain for a long time unobserved beyond the horizon. In the Wilhelmshaven experimental boiler some coals burnt almost without smoke, while others shewed a thick black smoke even twelve minutes after firing up. Three grades, expressing the thickness of the smoke, were adopted and in col. 12 of the table (p. 212) Smoke.

a signifies thick black smoke,
b „ transparent smoke of a grey or brownish colour,
c „ thin, scarcely visible smoke.

For warships, no coals are considered suitable but those which shew a transparent grey smoke for at the most, four minutes after firing. Most of the superior Westphalian coals fulfil these requirements, and those of fast mail boats, as well as Welsh coals. Ordinary English and Scotch coal smokes for a very long time; the smoke is however often of a lighter colour than that of the heavier coals which only smoke for a short period.

7) f. **Easy inflammability and rapid combustion** are desirable, so that, under certain circumstances, steam may be got up with extreme dispatch. Such a case occurs on the outbreak of fire on board a steamer when the boilers are not in steam and it is necessary to set the steam fire-engines to work in the shortest possible time. In warships similar and much more numerous cases may arise, during hostilities, in which the safety of ship and crew depend upon the promptness of raising steam. The rapidity of combustion is invariably greater in light than in heavy coals, and in the latter kinds it is improved by the more or less complete absence of small. Under similar circumstances, rapidity of combustion depends entirely on the strength of the draught and is therefore always greater with artificial than natural draught. Rapidity of combustion.

8) *The measure of the rate of combustion of the coals* is the weight which can be burnt per □m of grate surface per hour. Ceteris paribus this weight of coal may be taken as a scale of the Scale for comparison of the performance of the stokers, or of the forcing of the fires.

capacity of the stokers, seeing that a skilful fireman can, in a given time, completely burn — not simply throw on the fire — a much greater weight of coal than an unpracticed one. But with equally good firemen, the weight burnt per □m of grate per hour gives a comparison of the extent to which the boiler is forced, as a hard-fired boiler produces more steam than a moderately fired one. This number (of kg burnt per □m per hour) is therefore sometimes called the "degree of forcing" of the fires.

Average values of the degree of forcing in marine boilers.

9) The average weight burnt per □m per hour in ordinary return-tube boilers may be stated at

a) with natural draught

with *slow*	firing	40	to	60	kg of coal,	the boilers being	*worked easy*,
„ *ordinary*	„	60	„	80	„ „	„ „ „	*moderately*,
„ *sharp*	„	80	„	100	„ „	„ „ „	*driven*,
„ *very hard*	„	100	„	120	„ „	„ „ „	*greatly over-worked*;

b) with forced draught

at moderate pressure (up to 30 mm of water) 120 to 150 kg of coals, the boilers being *driven to the utmost*,

at heavy pressure (above 50 mm of water), 150 to 180 kg of coals, the boilers being *at the limit of their capacity*.

In cylindrical boilers of the "Navy" type, with forced draught at a heavy pressure, as much as 200 kg may be burnt per □m per hour, and in locomotive-boilers under ordinary conditions of forced draught, up to 300 kg. The utmost limit of rapidity of combustion has been found by experiments with locomotive boilers at the strongest draught (150 to 200 mm), to be about 600 kg per □m per hour. The average consumption of marine boilers on extended voyages and not overdriven is from 70 to 75 kg per □m of grate per hour.

Weight of coal for laying fires and banking.

10) *For laying fires* about 100 to 120 kg per □m are required on an average, and for keeping up the fires, when they are banked, about 5 to 8 kg per □m per hour must be reckoned.

Caking.

11) g. **A slight tendency to cake** is a quality to be sought in the coals, to keep them from slipping on the grates as the vessel rolls, thus exposing portions of the bars and admitting cold air, injurious alike to the process of combustion and to the boiler itself. A strong tendency to cake is, on the other hand, undesirable, as in consequence of it the air-spaces become obstructed and the difficulty of working the thick fires, necessary with artificial draught, is considerably increased.

12) **II. The suitability of the different kinds of coals for marine purposes** may be ranked as follows. **Suitability of the various kinds of coals for marine purposes.**

a. Bituminous, long-flaming coals are distinguished by great hardness and tenacity, so that they make but little small during the processes of transit and shipment. They are easily ignited and burn very briskly but with a great amount of smoke. On this account they are unsuitable for boilers of the marine type, having comparatively contracted furnaces. **Bituminous coal not adapted.**

b. Semi-bituminous short-flaming coals (Fettkohlen), which include many of the superior Westphalian descriptions, are the most used for marine boilers. The slightly-caking kinds are very suitable, whereas those more inclined to cake are not to be recommended on account of the constant pricking they require. All semi-bituminous coals burn easily under natural draught, but they smoke more or less, unless special smoke-preventing appliances are used. The best coals of this description possess sufficient hardness accompanied by high calorific power, leave but little ash and clinker, and being only moderately inclined to cake, require only slight attention on the part of the stoker to burn them with sufficient economy. **Semi-bituminous coal suitable.**

c. Slightly bituminous, short-flaming coals (Esskohlen), in which Welsh coals must be included are the best adapted for marine boilers. That they are nevertheless not so much used as semi-bituminous coals is due to their price being mostly too high for ordinary cargo steamers. The superior "Esskohlen" are also easily ignited, burn with a bright flame, do not cake much, smoke but slightly, and leave little ash and clinker. They are therefore comparatively easily worked on the grate, but they crumble when stored for long in the open air, or exposed to rough treatment in transit. Their objectionable tendency to form small is aggravated by their splintering during combustion so as produce a loss of 15 % under some conditions. The high commercial value of these coals arises from the slight care and attention they require to burn them economically and without smoke. **Slightly bituminous coal the most suitable.**

d. Non-bituminous short-flaming anthracite is used in North America for marine boilers, but is not to be recommended. Owing to its want of bitumen, it burns like coke almost without flame or smoke and with very intense local incandescence and requires besides a powerful draught, a high furnace temperature, and therefore great attention of the fireman to consume it economically. **Non-bituminous coals unsuitable.**

Table of coals used for steamers*).

Land	Name	Water evaporated (from 0°) per kg of coal per hour	Water evaporated (from 0°) per □ m of grate per hour	Weight per cbm broken in kg	Tenacity percentage	Percentages of unburnt residue: Clinker	Ash	Soot and ashes in smokebox	Together	Duration of smoking in minutes	Degree of thickness of smoke	Coal burnt per □ m of grate per hour in kg	Behaviour on the grate	Remarks.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Westphalia	*"Fettkohle"* Victor	9,277	854,76	730,00	45,20	0,54	2,25	0,34	3,13	1—2	b—c	92,14	caking	Easily ignited, burn with long bright flame and little smoke
,,	,, Recklinghausen	9,256	841,45	769,90	52,00	1,74	1,27	0,32	3,33	3—4	b	90,91	,,	,,
,,	,, Prinz Regent	9,052	874,19	760,00	47,00	1,42	2,09	0,34	3,85	3,00	b—c	96,57	,,	,,
,,	,, Holland	8,924	863,37	732,00	46,40	1,78	2,13	0,33	4,24	3,00	b	96,74	,,	,,
,,	,, Verein. Präsident	8,858	738,27	753,20	47,86	1,75	2,61	0,36	4,72	2,25	c—b	83,34	,,	,,
,,	,, Wolfsbank	8,756	785,54	767,80	47,74	2,43	1,84	0,36	4,63	2,00	b	89,71	,,	,,
,,	,, Shamrock	8,724	741,45	754,00	44,50	1,91	1,93	0,29	4,13	1,50	c—b	84,97	,,	,,
,,	*"Esskohle"* Franziska	8,403	794,83	767,00	47,28	3,74	4,38	0,25	8,37	7,00	c	94,60	non-caking	,,
,,	,, Caroline	8,333	720,17	762,00	51,64	1,93	5,89	0,36	8,18	nearly 0	—	86,42	,,	,,
,,	,, Ibbenbüren	8,372	765,34	753,90	35,35	1,76	5,37	0,41	7,54	1—2	,,	91,42	,,	,,
,,	*"Patent-Fuel"* Blankenburg	9,164	887,52	1080 ¹)	61,00	3,79	2,75	0,35	6,89	2—3	c	96,84	caking	,,
,,	,, Neu-Iserlohn	9,008	801,71	1080 ¹)	59,35	2,36	2,00	0,28	4,64	1—2	c	89,00	,,	,,
,,	,, Franziska Tiefbau	8,826	815,45	1100 ¹)	62,31	2,85	2,73	0,30	5,88	0—1	c	92,38	,,	,,
,,	Wurm-Revier. Aachen	8,776	757,39	766,00	58,20	1,07	2,64	0,44	4,15	10—12	c	86,31	non-caking	Allied to "Esskohlen"
Wales	Nant Melyn Merthyr	8,170	646,31	727,00	48,24	2,22	7,64	0,34	10,25	4—5	b	79,09	,,	,,
,,	Thomas's Merthyr	8,569	730,71	733,20	49,84	5,13	4,52	0,37	10,02	5—6	b	85,28	,,	,,
,,	Nixon's Navigation	8,412	921,99	744,66	48,80	1,21	4,07	0,32	5,60	5—6	b	109,60	,,	,,
England	Newcastle	7,278	823,38	708,00	77,32	1,93	2,57	0,35	4,85	12—13	a	113,12	,,	,,
Scotland	Lochgelly	6,888	819,11	675,70	82,47	1,76	1,95	0,34	4,05	7—8	a	118,92	,,	More like gas-coal
Japan	Origin unknown	6,592	629,26	762,00	78,60	5,81	6,04	0,41	12,26	7—8	a	95,45	,,	Very "fast" and smoky like gas-coal, long flame, moderate smoke
Australia	Gulland Mine. Brisbane	6,764	598,90	887,10	—	2,88	15,63	0,47	18,98	6—7	b—c	88,54	slightly caking	
,,	Wallsend Mine. Sydney	7,326	679,43	812,25	70,44	1,88	6,06	0,43	8,37	9—10	b	92,74	,,	,,
New-Zealand ²)	Bay of Islands	7,136	869,05	—	64,60	2,00	2,47	0,61	5,08	5—6	b	121,31	,,	,,
So. America	Punta-Arenas	4,474	546,40	818,60	—	13,64	8,12	0,29	22,05	4—5	b—c	122,12	non-caking	Quite useless for naval purposes

¹) This weight refers to patent fuel, packed, not broken.
²) New-Zealand coal from Westport, which is reported to be superior to the best Newcastle, was used on board the "Calliope" when steaming out of the Bay of Apia, Samoa, during the hurricane of March 16th. 1889.

*) Supplement to the Marine-Verordnungsblatt 15. October 1876. No. 19. — 28. February 1878. No. 4. — 15. February 1879. No. 3. — 15. May 1883. No. 10. -- 30. November 1886. No. 23.

13) **External signs,** affording any inference as to the quality of coals are extraordinarily difficult to distinguish, so that only the following can in general be said on the subject. Flaming coals, allied to lignite, are comparatively hard, possess some ring, are tough, of uneven fracture, and in colour of a dull black inclined to brown. With decreasing oxygen the coal becomes blacker and denser, loses its ring, and is more easily pulverized. The lustre and the tendency to cake increase directly as the proportion of hydrogen. Coals of an anthracitic character are as rule pure black and a little softer than the bituminous short flaming kinds. Density and hardness increase with the percentage of ash, while the lustre diminishes. However all the above qualities of the different kinds of coal are modified by earthy impurities when these are present. **External characteristics of coals.**

14) *As a criterion of the goodness of coal*, when buying it at a foreign port and in the absence of any certificate as to quality, only the following rules can be given. The coals must not be "perished" by exposure to the air, nor wet, nor "brassy", nor accompanied by much small (§ 23. 14). They should have a bright black appearance, and yield sharp bright dust on being broken. The more the colour is inclined to brown or grey, the fracture slatey and the dust earthy, the lower is the value of the coals. **Criteria of the goodness of coal.**

15) The table on the preceding page contains the particulars of the most important kinds of coal got in various countries and used for marine boilers, as determined by the Engineering staff at Wilhelmshaven Dockyard during a course of experiments extending from 1874 to 1886. The experiments have incidentally borne witness to a satisfactory extension of the output of the superior kinds of Westphalian coal of late years and to considerable progress in the, at present, flourishing manufacture of patent fuel. **Table.**

§ 23.

The stowage and supervision of fuel on board.

1) On steam ships, the fuel is stowed in spaces, as near the boilers as possible, called the *bunkers*. The arrangement and fitting of these spaces is different according as they are intended for the reception of solid or liquid fuel (petroleum refuse). **Bunkers.**

Arrangement of bunkers.

2) **I. Stowage of coals.** **The arrangement of the bunkers** depends upon the space available, the stability and trim of the ship, and in vessels of war upon certain considerations of the defence of the machinery and boilers. They are generally planned as follows —

One bunker forward of the boiler space,
„ „ in each wing in the boiler space,
„ „ on each side between the boiler space and engine-room.

Sometimes the bunkers extend along one or both sides of the engine-room. Recent iron-clads and cruisers have cellular spaces filled with coals over their armoured decks, as a protection. The height of the bunkers depends upon the size of the ship; they may extend either to the lower, main, or upper deck. Figs. 11 and 12, Plate 5, shew the bunker plan of the Hamburg Mail Steamer "Columbia", and Figs. 5 to 10 that of a cargo steamer.

Bunker bulkheads.

3) **The bunker bulkheads** (in wooden ships also) are composed of 6 to 13 mm plates stiffened with tee or angle irons and stayed to the ships side. In war ships and high class merchant vessels portions of the bunker bulkheads are made watertight.

Bunker doors.

4) **Communication** between the bunkers and the boiler room is usually obtained through sliding doors, which in the case of watertight bulkheads can be closed in time of danger from one of the upper decks (Plate 6, Fig. 18). On Plate 7, Figs. 1 to 4 shew a large, and Figs. 5 to 10 a small pattern of such doors. Both of them are opened from the stokehole by means of a crank, wormwheel, and rack. If they have to be closed suddenly, a loose clutch, connecting the pinion of the rack with the worm-spindle, is thrown out of gear from the upper deck and the door falls. The bronze wedge-shaped lugs on the door engage with suitable provisions on the frame and force the door home against the frame perfectly watertight. Persons passing through the openings at the time of closing the doors incur immediate risk of their lives. Effects have recently been made to devise slow-closing doors, the most successful one being MAC ELROY'S*), as now fitted on a large number of steamers. This door (Plate 7, Figs. 11 to 17) has, instead of the rack of the ordinary door, a screwed spindle which gears into two bronze worm-wheels as in a nut (Fig. 16). So long as these wheels are held in one position the door can be opened or closed from the stokehole, by means

Mac Elroy's watertight door.

*) English Patent No. 3610. 1886.

of the handle. The wheels are kept stationary by steel brake bands controlled by a lever (Fig. 15). The free end of this lever engages in a wrought iron nut, working upon a short bronze screw (Fig. 14). This screw can be turned but cannot move end-wise. If it is turned in one direction the nut travels upwards, raises the lever, slackens the brakes, allows the worm-wheels to revolve, and the door to sink by its own weight; and if the screw be turned in the reverse direction the brakes are tightened, the wheels held, and the door arrested in its momentary position. Through this short bronze screw passes a square rod, having one quarter of a turn of twist in its length, and extending to the upper deck. This rod is held in a certain position by a stiff spiral spring (Fig. 11). By turning the rod with a suitable spanner on deck the worm-wheels can of course be held or released. If it is turned in the direction to close the door, the hollow screw sliding on the twisted rod is caused to revolve in such a manner as to keep a certain tension on the brakes, while the spiral spring (Fig. 13) exerts a continuous effort to turn the square rod back into its position of rest, thus assisting the slow closing of the door. By suitably turning the twisted rod with the spanner on deck, the door can either be suddenly stopped or merely retarded in its downward motion.

5) **The filling of the bunkers** is accomplished by means of either **Filling the bunkers.**

a) Hatches,
b) Ports,
c) V Shoots, or
d) Elevators.

6) a. **Hatches** (Plate 6, Figs. 1 to 8) are fitted on wooden steamers, on iron and steel steamers of low free-board, and on warships having side-armour, in all of which the coals must be dropped into the bunkers from above. The hatches are surrounded by cast iron rings of about 40 cm clear diameter. These rings are let into the deck plank and fastened with wood screws and are provided with a ledge on which rests the cover of cast iron (Figs. 7 and 8) or, when greater durability is required, of wrought iron (Figs. 1 and 2). The joint is made by smearing the ledge with tallow, on which the cover presses. When the hatches are placed over inhabited quarters or alleyways, the covers are generally fitted with dogs or other arrangements for screwing them up tight, and are often provided with glasses for admitting light. Those communicating direct with the bunkers are kept closed by their own weight or, in rare cases, are fitted with a bayonet joint. **Hatches.**

Coaling trunks and plates.

7) If the coals have to fall through one or two decks before they reach the bunkers (Pl. 6, Fig. 18), the several coaling hatches which are over each other are connected before bunkering by two sheet iron cylinders fitting one on the other, called *coaling trunks*, which are hung up beneath the upper deck, as shewn in Figs. 11 and 12. To protect the upper deck from the coals, iron plates, called *coaling plates* are placed round the hatches (Pl. 6, Figs. 18 and 20).

Bunker hatch gratings.

8) In order to ventilate the bunkers, the covers of the hatches are from time to time exchanged for *gratings*. The best place to carry these gratings when the covers are on is in the hatches themselves beneath the covers, as shewn in Pl. 6, Figs. 1 and 9. When the gratings are to be used, the covers are taken off, the gratings lifted and fitted with their lugs into suitable recesses in the rings, so that they lie flush with the deck (Fig. 3). A neater arrangement is shewn in Figs. 7 and 8, by which the grating only requires to be turned over to bring it flush with the deck. For the better ventilation of the bunkers, as described in 19) the bunker hatch ring is often fitted to receive the foot of a revolving cowl ventilator (Pl. 6, Figs. 9, 10, and 19).

Coaling ports.

9) b. **Coaling ports** are used on high-freeboard iron and steel ships, large mail steamers, and cruisers. They are cut in the ship's side a little above the water-line and are closed either with loose plates bolted on, or doors secured on the inside. The joint is made either with red lead, indiarubber, or felt, and during coaling the india-rubber or felt is protected by thin strips of plate screwed on over it. Sheet iron shoots are placed in the *coaling ports* (Pl. 6, Figs. 15 to 17), so that the coals may be tipped straight into the bunkers, either from railway trucks or lighters. The process of coaling with ports is more rapid, handy, and cleanly than that described above.

V shoots.

10) c. **V Shoots** are now often fitted on British cargo steamers (Pl. 5, Figs. 5 to 10). These vessels lie immediately alongside the high spouts in the British coaling ports, and the railway trucks empty their contents into a trunk hatch standing above the upper deck, placed between the engine and boiler spaces, and having a guide or "shoot" of a saddle or inverted V form beneath it. The coals slide down the inclined surfaces of this shoot into the side bunkers, the sides of which are also of a sloping shape at their upper parts, and the bunkering is thus very rapidly accomplished. The tween-deck bunkers, above the shoot, are filled by means of small hatches placed on the upper deck for this purpose.

11) d. **Coal elevators** *), which are used in several British colonies for discharging coal cargoes, have been proposed by RIGG for bunkering steamers in order to reduce the expense and, what is still more important, to shorten the time required by the ordinary methods in large vessels. As there is as yet no experience of this system, it is uncertain whether the coals would not be fed into the bunkers faster than they could be properly trimmed. The much-discussed question of coaling war-ships at sea still remains open, as the elevators could only work in smooth water. Coal elevators.

12) *A temperature tube* should be fitted in each bunker. This is a wrought iron tube (Pl. 6, Fig. 18), starting from one of the upper decks and carried down through the bunker nearly to the ship's bottom. It is closed at the top with a deck plate and serves for introducing a thermometer into the bunker, to ascertain its temperature. It is advisable to label the deck plate so as to distinguish its purpose, as shewn in Fig. 14, Pl. 6. Temperature tubes.

13) **II. Supervision of the coals.** The coals, when stowed in the bunkers, require constant supervision, as they are liable to spontaneous ignition, and sometimes form explosive gases. The British Government has had the causes of spontaneous ignition and formation of gases in coal stowed in bunkers inquired into by a Royal Commission. The essential points in the Report **) of this Commission are the following. British Commission of Enquiry.

14) **The spontaneous ignition of coals** is brought about by two principal causes — Causes of spontaneous ignition.

a) Chemical changes of the coals,
b) Their capacity for absorbing gases.

15) a. *Chemical changes* are undergone by certain constituents of coal, under the influence of atmospheric oxygen. Of these constituents, sulphur and iron, as pyrites, occur most frequently. Moist air assists the oxydising of pyrites by bringing more oxygen into contact with the surface of the oxydisable body. As every oxydation is accompanied by a liberation of heat, the temperature of the coals is gradually raised. But the higher the temperature gets, the more active the oxydation becomes, so that these two causes react upon each other until the ignition temperature of the coal is reached. This process Chemical changes of iron pyrites.

*) Journal of the united service institution. 1889. P. 977.

**) Report of the Royal Commissioners appointed to enquire into the spontaneous combustion of coal in ships. London 1875.

is accelerated by the comparatively high temperature of the stokehole and also by the heat of the tropics.

Condensation of gases in porous coals.

16) b. *A pronounced capacity for absorption* is possessed by very "small" or porous coals for certain gases, particularly oxygen. Some heat is always produced when a gas is absorbed or condensed in a porous body. The tendency to oxydation of the coals is increased by the condensation of the gases in their pores, because the combination of the carbon with oxygen is assisted thereby, so that absorption and oxydation are both generating heat at the same time. As the temperature rises so much heat is produced that the coal is ignited by the chemical action. Moisture is an obstacle to the latter as it fills up the pores and hinders the absorption.

No special ventilation.

17) According to the views of the Royal Commissioners, coal stowed in the bunkers will, on tropical voyages, always generate a certain degree of heat, to overcome which no practicable system of ventilation would suffice. On the contrary, the continued introduction of fresh oxygen by ventilation among the coals, would only favour the oxydation, thus causing such a generation of heat as would be highly dangerous.

Protective measures against spontaneous ignition of coals.

18) *Means for preventing spontaneous ignition* of coals are

a) not to ship certain "brassy" or pyritic coals, at all;
b) nor any coals containing much small or in a wet condition;
c) to ascertain the temperature of the bunkers by means of the temperature tubes regularly every watch when under steam, and every twelve hours when under sail.

Explosion of coal gases.

19) **Explosions of coal gases** occur more rarely than spontaneous ignition. They arise from coals out of pits containing marsh-gas being shipped fresh. The marsh-gas escapes from the coals in the closed bunkers, mixes with air, and forms an explosive mixture. It however never explodes spontaneously, but only when it is ignited by an open light.

Preventive measures against coal-gas explosions.

20) *Measures for preventing explosions of coal-gas* are

a) Surface-ventilation of the coals, giving the gases as they are disengaged the freest possible escape. This is accomplished by shipping the bunker hatch gratings, which should be done at least twice a week:
b) The strict observance of the rule that bunkers, which have been closed for any length of time, are only to be entered with safety-lamps, such as are used in collieries. Lamps of this description are always provided for vessels of the Imperial German Navy and should never be wanting in merchant steamers.

21) *The extinguishing of coals which have ignited spontaneously in the bunkers*, by means of carbonic acid, as has frequently been proposed, is not considered advisable by the Royal Commissioners, as this gas has no cooling influence. They are of the opinion rather, that only water or steam can be effectively used for putting out fires which may have arisen in considerable masses of coal. To avoid the flooding of large bunkers in such cases, they have often been placed in connection with the boilers by means of steam pipes. In case of fire breaking out, the stop-valves on these pipes are opened, and the steam rushes in among the burning coals. Putting out fires in bunkers.

22) As, in spite of these regulations, the number of cases of spontaneous ignition of coals and of explosion of gas have only very slightly diminished since 1875, ROWAN*) has proposed the following additional clauses: Rowan's proposals.

a. At different positions among the coals some mechanical appliances of the nature of ejectors are to be placed, which can be made to draw out the warm gases, thus preventing spontaneous ignition, in case an increase of temperature threatens an approaching chemical change in the coals.

b. Before the coals are shipped they are to be exposed to a temperature of about 65° C. in a building provided for the purpose during a period varying from 24 hours to 7 or 8 days, according to their nature, and are then to be cooled down and protected from moisture. By this process all gases and moisture absorbed by the coals are driven off, so that the possibility of their subsequent oxydation is greatly diminished, but the attendant expense is so great that there is little prospect of its general adoption. Drying the coal.

23) **III. Stowage of liquid fuels.** The best guides for the stowage of liquid fuels on board ship are the regulations**) published by the Russian government as to the firing of steamers with petroleum refuse on the Caspian Sea, as well as the arrangements***) adopted on board the steamers recently built for carrying petroleum in bulk instead of in barrels****). The following are the leading Stowage of oil in bunkers.

24) **Points to be observed in designing bunkers for oil.** Whereas oil was formerly always carried in separate vessels, the weight of which General experience.

*) TH. ROWAN. Coal. Spontaneous combustion and explosions. London 1882.

**) ST. GULISCHAMBAROW. Neftjanoe otoplenje parochodow i parowosow. (The use of naphtha as fuel for steamers and locomotives.) St. Petersburg 1883. P. 135.

***) Engineering 1886. II. P. 107.

****) Zeitschrift des Vereines deutscher Ingenieure. 1886. P. 1083.

was considerable, the experience gained with modern tank steamers for the petroleum trade shews that liquid fuels can be carried in bunkers just as well as coals if proper attention is paid to

a) The expansion of the oil under increases of temperature,
b) The losses of oil through leakage at the seams,
c) The separation of the water often contained in the oil,
d) The influence of partially emptied bunkers upon the vessel's stability,
e) The removal of gases generated from the oil,
f) Keeping the oil at a proper distance from the stokeholes,
g) Warming the oil in time of severe frost.

Expansion.

25) a. *The expansion of the oil*, on an increase of temperature, due either to radiation from the fires or the heat of the tropics, is very considerable. According to ST. CLAIRE DEVILLE'S investigations, the coefficient of expansion for the different kinds of petroleum varies from 0.0007 to 0.0009, so that a rise of temperature of 22^0 to 23^0 C. is sufficient to cause an expansion of 2 $^0/_0$ of the bulk of the oil. Therefore, in order to keep all the bunkers full, except just the one being worked from, they must be provided with trunks or similar means for allowing of the expansion and contraction of their contents.

Losses.

26) b. *The losses* arise from the peculiarity of the oil to leak through joints and seams which are perfectly tight against water. This tendency however, decreases with the limpidness of the oil, so that it is weaker in petroleum refuse than in refined petroleum. According to experience the loss from leakage of the latter in barrels is about 2 $^0/_0$ of the gross weight. In bunkers where of course there is a much shorter length of seam per unit volume of the oil than in barrels, this loss, with careful riveting, is found to be much smaller. It therefore appears probable that in bunkers filled with petroleum refuse the leakage would hardly amount to 1 $^0/_0$.

Separation of the water.

27) c. *The separation of the water contained in the oil* must be effected during the time it remains in the bunkers, in order to avoid interruptions of the firing, such as steaming and sudden extinction of the flame, and also to prevent losses of heat. This water becomes mechanically mixed with the oil, partly from dew, rain &c. during transit in railway trucks, and partly from external leakages of lighters, &c.

Effect upon the ship's stability.

28) d. *The effect upon the ship's stability* when cross-bunkers, especially large ones, are partly emptied, may become positively dangerous. Assuming a cross-bunker to be so far emptied as

to allow the oil to flow over from the emerged side OHS, to the immersed side $O_1 H_1 S$, of the ship when she is inclined, (see Fig. 3, Pl. 5) then we not only have the centre of buoyancy shifted from B to B_1, but also the centre of gravity of the ship shifted from G to G_1. The line of buoyancy passing vertically through B_1 cuts the centre line of the ship in M, and the vertical line of gravity passing through G_1 cuts the centre line in A, so that AM is now the measure of the righting moment. This effective metacentric height is shorter, by the distance GA than it is when the bunker is quite full and allows of no alteration in the position of the ship's centre of gravity. The metacentric height is still further reduced (see Fig. 4, Pl. 5) when the ship has under the bunker a double bottom or an empty ballast tank, into which the oil may leak in considerable quantities and run down to the immersed side.

29) e. *The removal* of the gases disengaged from the oil by heating or concussion, and collecting on the surface of the oil in a bunker which is being worked from, is absolutely necessary in order to prevent risk of fire or explosion. **Removal of gases.**

30) f. *All leakage through the bunker bulkheads must be kept clear* of the stokeholes for the same reasons as above. Besides which, the pungent and extremely unpleasant smell of the oil leaking out would be intolerable for any length of time to the engineers. **Contact with external bunker bulkheads to be prevented.**

31) g. *Warming the oil* is necessary during hard frost or in high latitudes, in order to keep it sufficiently limpid for burning in the jets; it must be begun when the temperature of the air reaches — 12° C. The process of warming separates the water and any other coarse impurity. **Warming the oil.**

32) **Construction and fitting of oil-bunkers.** From the above considerations and from the Russian rules, we may gather the following suggestions for the construction of oil bunkers. **Arrangement of oil bunkers.**

33) α. Wood must not be used either for the bulkheads, bottoms, or covers, as it is too dangerous. **Wood excluded.**

34) β. New ships, to be fitted originally with oil bunkers, must have double bunker bulkheads throughout in way of engine and boiler spaces. It is advisable to keep the intermediate space always filled with water, or at least to fit a collector at the lowest part of the space so that any oil leaking through the bunker sides may accumulate there and be withdrawn from time to time by means of a special pump. Any gases which form in this space (assuming that it is not filled with water) are driven off by an ejector. Neither suice-valves nor **Double bulkheads.**

any other openings must be fitted in the insulating bulkhead. In short, everything must be done to prevent the possibility of oil leaking into the engine and boiler room bilges.

Shell riveting.

35) γ. In new ships the shell and double bottom can form the boundaries of the bunkers, provided that in way of these they are riveted like boilers and are tested with cold water to twice the pressure they will have to stand when at work. Sir WILLIAM ARMSTRONG & CO'S. oil ships, designed by Mr. H. SWAN, are constructed in this manner*). The following are the particulars of riveting &c. of an oil ship built by Messrs. WILLIAM GRAY & CO. of West Hartlepool**). The inner bottom plating which formed the lower boundary of the oil-space, was 15 mm thick fastened with 16 mm rivets spaced 57 mm apart. In the shell-plates, which were 16 to 17.5 mm thick the rivets were 22 mm diar., spaced 63.5 to 70 mm in the edges and 70 to 76 mm in the butts. Where 19 mm rivets were used they were spaced 60 to 63.5 mm apart. The shell rivets in the frames were spaced 152 to 165 mm. The laps of bulkhead plating had 16 mm rivets spaced 57 mm. This close spacing, combined with good fitting of the plates, holes drilled in place, and countersunk on both sides, careful riveting and caulking, secures the practical oil-tightness of the work.

Decks over bunkers.

36) ε. The decks, in way of bunkers must be of iron, with the same oil-tight riveting as in the shell and double bottom. If the iron deck is to be sheathed with planking, it is advisable to cover it first with a layer of felt or cement and to coat the deck-planks on their sides and faying surfaces with glue or varnish, to protect them as far as possible from becoming impregnated with oozing oil. The risk of fire is however diminished by omitting wood sheathings altogether. In place of the ordinary coaling hatches, well-jointed manholes are to be fitted, giving access to the bunkers.

Fitting coal bunkers for the reception of oil.

37) ζ. Bunkers which have been used hitherto for coals will not hold oil, as they are not sufficiently tight. They must be lined throughout with oil-tight bulkheads, forming in fact a series of carefully constructed tanks built into the bunkers and conforming as far as possible to their shape, to avoid waste of space.

Automatic filling arrangements.

38) η. All bunkers must be arranged to fill themselves up automatically, so as to remain completely full till their contents are wanted. As already mentioned, the oil contracts to a

*) Engineering. 1886. II. P. 114.

**) Ibid. P. 111.

certain extent when the temperature falls, and there is besides a constant leakage of oil through the ship's skin into the sea, although no water can penetrate the oil-tight seams and butts. In bad weather, the oil in a bunker thus partially emptied oscillates considerably, tending of course to increase the extra strains upon the ship; the fastenings are more or less disturbed and the leakage augmented. This difficulty is met by keeping the bunkers constantly full, which is effected by connecting them all together, or perhaps better separately, with an oil-holder placed higher up in the ship which serves not only to fill up the spaces emptied by leakage or contraction but also to take up any increase of volume due to expansion under increase of temperature. It is obvious that the smaller the level surface of the oil in this holder can be kept, the less it will affect the ship's stability. The best arrangement is a separate cylindrical tube or trunk of sufficient capacity fitted at the stop of each bunker and extending to the next deck.

39) ϑ. Cross-bunkers must be divided by longitudinal bulkheads into two or more compartments, so as to affect the stability as little as possible through changes in the oil-level when the ship is heeled over. **Longitudinal bulkheads for cross-bunkers.**

40) ι. Each bunker must be connected at its highest point with a ventilating pipe leading to a cowl on deck which may either be separate for each bunker, or fitted with branch pipes to several. The mouths of the cowls must be covered with wire gauze, to prevent the escaping gases from catching fire. Each bunker must also be provided with a filling pipe. **Ventilators.**

41) $\varkappa$. At the lowest point of every bunker there must be a collecting-box for the water accumulating from the oil, connected either with the bilge-pumps or an ejector. **Collecting boxes.**

42) λ. The bunkers of steamers navigating high latitudes must be provided with steam coals for warming the oil and separating the water mixed with it. **Steam coils.**

43) μ. The boilers must be carefully cleaded and there must be a clear space of at least 45 cm between them and the bunker bulkheads, to prevent heating the oil. **Distance from the boilers.**

44) ν. The bunkers are to be fitted at their lower parts with gauge glasses, to shew when they are emptied. **Gauge-glasses.**

45) **IV. Supervision of the oil.** The frequent occurrences of ignition of the oil in the bunkers of steamers, particularly in the Caspian, have often been ascribed to spontaneous combustion, whereas they are really induced by the easy inflammability of **Inflammability of the oil.**

the oil and reckless handling of it. According to the researches of BUTLEROW and SININ, spontaneous combustion of petroleum is impossible, because it condenses no gases on its surface nor do any of its chemical components take up oxygen from the air, all of them in fact strongly resisting any high degree of oxydation. Spontaneous combustion can only occur in the presence of finely divided substances, such as sawdust, cotton-waste, and the like, completely soaked with petroleum. In order to put an end to the danger arising from the easy inflammability of the oil, the Russian government has prohibited the use of any petroleum or petroleum refuse as fuel for steamers, the flashing point of which is below 70° C., because the temperature of the oil in the bunkers or holders may rise above 60° C. through radiation from the boilers and the heat of the sun.

Use of crude oil. 46) This veto only affects the application of crude oil as fuel, whereas the residue of the first distillation of this oil (into illuminating oil), like the other heavy oils used as fuel, has a flashing point of about 100° C. GULISCHAMBAROW is however of opinion that crude oil could be used as fuel so long as due care were taken with regard to the evaporation of its volatile parts (see § 19. 16); and the soundness of this view is shewn by the North American experiments, where crude oil is almost exclusively used.

Precautionary measures. 47) The precautionary measures to be observed in using petroleum as fuel are all based upon the single consideration that there is nothing to fear from the oil itself, but only from the gases volatilized from it and forming under the access of air an explosive mixture, which however does not ignite except on contact with a naked flame. If, therefore, the oil-bunkers are fitted with proper pipes for carrying off these gases, are also carefully ventilated after emptying, and then only entered with safety-lamps, the use of petroleum refuse, giving off scarcely any gases, (instead of using crude oil) is not only absolutely unattended with danger, but is positively safer than the use of coals, subject as they are to spontaneous combustion. But of course if naked handlamps are held inside half-empty bunkers, which piece of incomprehensible recklessness is reported of Russian engineers, serious consequences must ensue.

Engine-room instructions. 48) All Russian steamers using oil as fuel are obliged to carry a notice-board in the engine room, calling attention to the dangers arising from the use of naked lights and from entering the bunkers before they are thoroughly ventilated. The

Russian government engineers are responsible for the unconditional compliance with all regulations referring to liquid fuel on the part of those in charge of these steamers.

49) **V. Measurement of the bunkers.** The contents of the bunkers are calculated, like all spaces in ships, by SIMPSON'S rule. As the bunkers on both sides are almost always arranged symmetrically with regard to the midship fore-and-aft plane of the ship, the measurement of the one side bunker and half the thwartship bunker is generally sufficient for determining the quantity of coal that can be stowed. It is most convenient when the lengths of the side and thwartship bunkers contain an even number of frame spaces, or as is shewn in Figs. 13 and 14, Pl. 5, an even number of double frame spaces. If, however, this is not the case, the length of the bunkers must be divided into an even number of parts, without regard to the frame spaces. **Handiest method.**

50) In order to calculate the area of each of these sectional planes, the height h (in metres) is measured with a tape and marked, in side bunkers on the bunker side and in cross-bunkers amidships. It is best to divide this height into $n = 10$ equal parts and mark them on the bunker side or on a vertical batten. The breadths or horizontal ordinates x_1, x_2, x_3, &c., are then measured with another batten and a level. By SIMPSON'S rule the area of the section then is **Calculating the cross-sections.**

$$i = \frac{h}{3\,n}\left[x_1 + 4\,(x_2 + x_4 + x_6 + x_8 + x_{10}) + 2\,(x_3 + x_5 + x_7 + x_9) + x_{11}\right] \square\text{m}.$$

51) If all the sectional areas are different, which may sometimes happen in small vessels with fine lines, the different sectional areas must be summated by SIMPSON'S rule, to get the cubic contents $\mathcal{J}$ of the bunker. If l in metres is the horizontal interval of the sections i_1, i_2, i_3, and $m\,l$ the total length of the bunker, then **Calculation of the contents.**

$$\mathcal{J} = \frac{l}{3\,m}\left[i_1 + 4\,(i_2 + i_4 + i_6 + \ldots) + 2\,(i_3 + i_5 + \ldots) + i_{m+1}\right] \text{cbm} \,.\,.\, (107)$$

52) As a general rule the bunkers are in the neighbourhood of the midship frame, and in ships with a straight middle body, a number of cross sections are either equal, or so nearly equal that the differences do not affect the calculation. For instance, in the ship shewn in Figs. 13 and 14, frames 66 to 76 are all $= a$, frames 60 to 65 nearly $= b$, and frames 56 to 59 about $= c$. In such cases the result is more rapidly arrived at by regarding the space in way of equal, or nearly equal frames as a prism, the contents of which are equal to its sectional area multiplied by its length. **Shortening the calculation.**

Example.

53) The steamer (Fig. 14) has three bunkers, a thwartship bunker forward, one port and one starboard side bunker. The height of the cross-bunker measures $h = 6$ m, and after dividing this into 10 equal parts, the breadths are as shewn in Fig. 13, so that the area of the half section is

$$i = \frac{6}{3 \times 10}\left[5.70+4\,(5.71+5.75+5.73+5.70+5.00)+2\,(5.72+5.75+5.71+5.50)+0.00\right] \square \text{ m}$$

$$i = 32.52 \ \square \text{ m}.$$

The area of the whole cross-section is therefore 65.04 □ m, and as the cross-bunkers is 2.4 m long, its cubic contents are 156 cbm. The side bunkers are easy to calculate as far as station 66. Half the contents of two frame-spaces (1.2 m) of the cross-bunker $= \frac{156}{4} = 39$ cbm, and subtracting from this the contents of the prismatic space between 70 and 72, the base of which, $ABCD$, may be regarded as a rectangle, we get the contents of two frame spaces of the side bunker as

$$39 - (6 \times 3.5 \times 1.2) = 13.8 \text{ cbm}.$$

Statement of the contents of the bunkers.

54) It is extremely useful to calculate the capacity of the bunkers either per frame-space or per beam-space, or at any rate for every two of these spaces, to enable the engineers at any time to estimate with safety the stock of coal on board from the bunker-plan (Fig. 14). In most Navies it is the rule to supply such a plan to every ship. Fig. 13 shews the method of measuring the bunker sections b and c for getting out the plan. The height at station b is 5.5 m, and at c 5.2 m, so that the sectional areas are

$$i_b = \frac{5.5}{3 \times 10}\left[2.0+4\,(2+2.1+2.1+2+1.5)+2\,(2+2.1+2.1+1.9)+0.0\right] = 10.45 \ \square \text{ m}$$

$$i_c = \frac{5.2}{3 \times 10}\left[1.9+4(1.9+1.95+1.9+1.85+1.1)+2(1.9+1.95+1.9+1.7)+0.0\right] = 8.94 \ \square \text{ m}.$$

These give, per 2 frame-spaces (= 1.2 m) a capacity of

$$10.45 \times 1.2 = 12.5 \text{ cbm}$$

$$8.94 \times 1.2 = 10.7 \text{ cbm}.$$

Then by Fig. 14, the capacity of each side-bunker is 100.3 cbm, or both of them 200.6, which added to 156 cbm for the cross bunker, makes a total of 356.6 cbm. Assuming (see § 19, 13, P. 153) that only 1 ton of coal can be stowed in 1.3 cbm of bunker space, on account of the angle-iron stiffeners, the impossibility of trimming the coals right up to the beams, and the unavoidable interstices among them, these bunkers will hold $\frac{356.6}{1.3} = 275$ tons of coal. In the German Navy, on coaling a new ship for the first time, great atten-

tion is paid to the weighing and trimming of the coals, in order to get a comparison with the measurement of the bunkers. They are found to hold on an average 3 to 4 % more than the calculated quantity, according to the form of bunkers and the description of coal.

§ 24.

The use of liquid fuel in marine boilers.

1) As great efforts have of late again been made to introduce liquid fuel for steamers, especially torpedo-boats, not only in the petroleum countries, North America and Russia, but also in England and France in spite of former failures, — the following impartial comparison of its advantages and disadvantages may be of interest in many circles. Object.

2) **I. The advantages of liquid fuel as compared with coal,** especially in fast modern warships, may be enumerated as Advantages.

a) rapidity and cheapness of shipment,
b) stowage in otherwise unemployed spaces,
c) reduction of stoke-hole staff,
d) greater evaporative power,
e) more complete combustion,
f) better ventilation of the stoke-hole,
g) improved command of the machinery,
h) increased life of the boilers,
i) simplicity and exactness of measurement.

3) a. **The rapidity and cheapness of shipment** is so great that, according to TWEDDELL*), the steamers on the Caspian take on board 800 to 1000 tons of oil in 3 to 4 hours, which time could be shortened by the use of less inefficient appliances. A torpedo-boat which would have, at the outside, 20 tons of liquid fuel to take would be finished in a few minutes, and it would occupy no more time to supply a whole flotilla than it does now to coal a single boat. Rapidity of shipment.

4) b. **The stowage of the oil in such parts of the ship** as are not available for cargo or other purposes would be possible, for instance, the ballast tanks of cargo steamers, the cellular bottom space of iron-clads, and the space below the close ceiling before and abaft the engine-room in other vessels. Such an application Stowage between the floors.

*) Journal of the united service institution. May 1885. P. 698.

of these spaces, permitting as it would, a reduction of the ordinary bunker capacity, could be made conducive to greater accessibility of the machinery, particularly in small craft.

Fewer stokers.

5) c. **The diminution in the number of stokers** and in the labour of those still retained, follows as a matter of course from the fact that the oil is taken from the bunkers by a steam pump to the holders above the furnaces, whence it runs down to the jets; so that, to begin with, all trimming is done away with. Besides, only one fireman per watch is required in each stoke-hole, to regulate the jet-cocks and look after the water-gauges. According to TWEDDELL, the Caspian steamers carry only one fireman and two boys for each watch. The largest steamers, with communicating stoke-holes, would therefore only require one fireman and three or four boys per watch, in all 3 firemen and 10 to 12 boys, whereas they now carry 60 to 80 firemen and trimmers, costing the owners, at a low estimate, 60 l. a head per annum for wages and victualling. It is easy thus to estimate what sums the large steam-ship companies would save by the reduction of the stoke-hole staff. — In torpedo-boats, one fireman per watch would be sufficient instead of the two now usually carried, and as the work in the stoke-hole would only be easy, the health of the men in bad weather would be better then at present, when it is found by experience that the heavily worked stokers are the first to get knocked up.

Greater evaporative power.

6) d. *The greater evaporative power* of liquid fuels, which is in the ratio of about 7 to 4 to that of coals, enables steamers either to carry a smaller weight of fuel to steam the same distance, or to increase their steaming distance if the same weight of liquid fuel is taken as of coal. A modern torpedo boat for instance, having a coal-endurance of 4000 knots at a certain speed, would be able to cover 7000 knots with an equal *weight* of oil instead of coal, or if the same bunker *space* is filled with oil 7500 knots, because 1 ton of coals takes up 1.25 cbm, but 1 ton of petroleum refuse only 1.11 cbm. By the help of oil fuel the range of action of torpedo boats may therefore be nearly doubled.

More complete combustion.

7) e. **The more complete combustion** of liquid fuels hinders the formation of solid residue and smoke. As the combustion is without residue neither clinker nor ash is formed, so that there is no cleaning fires nor sweeping tubes. The former point is particularly important for torpedo-boats, which have mostly only one fire, and when coals are used this becomes dirty after 6

or at the most 10 hours' steaming at full speed. It is then absolutely necessary to clean it, if the pressure and speed are not to continue falling off. But the process of cleaning the grate, which on the latest torpedo-boats measures nearly 4 □ m and is placed low down, is not so soon completed, considering that about 12 tons of coal are burnt in the 10 hours, leaving under the most favourable circumstances, 3 to 4 kg of ashes. The boat thus suffers a sensible loss of speed in consequence of the retarded evaporation. The speed is further reduced in consequence of the formation at the tube-ends of *swallow's nests*, as particularly described later on under artificial draught. During a protracted full-speed run it becomes necessary to clean the fire at shorter intervals, because the clinker is never quite got rid of. It is thus evident that a torpedo-catcher, only requiring one of her fires to be cleaned per watch, thus suffering less reduction of steam, will, after several hours' chase, overhaul a torpedo-boat of somewhat superior top speed to her own, but having only one fire. If however, liquid fuel instead of coal is used on board the torpedo-boat, she can go on steaming at full speed for days and will probably escape.

8) It is unnecessary to enlarge upon the riddance from the very troublesome and dirty operation of getting the ashes overboard which is even hardly practicable in action, also from the sparks continually issuing from the low funnel when under a high air-pressure, of which every one who has passed any length of time on a torpedo-boat retains anything but an agreeable recollection. **Absence of residue.**

9) The absence of smoke is besides very valuable strategically. The torpedo-boat whose position is not betrayed by her smoke will be very difficult to detect on the horizon and will thus always have an advantage, because she can recognize all other steamers by their smoke. How voluminous and how treacherous the masses of smoke become, when forced draught is suddenly applied in order to increase the speed, was again emphatically shewn during the British Naval Manœuvres of 1887 as reported by the Times. With oil fuel, a thick impenetrable smoke can be produced by suddenly cutting off the air-supply, and in the opinion of Capt. CURTIS R. N.*) this circumstance might be made use of for transmitting smoke signals to very long distances by MORSE'S system. **Smokeless combustion.**

*) Journal of the united service institution. May 1885. P. 696.

Improved stoke-hole ventilation.

10) f. **The improved ventilation of the stoke-hole** is brought about by the powerful air-suction of the jets and the reduced radiation. TWEDDELL*) states that on entering the stoke-hole when the external shade temperature is 40° C. scarcely any increase of temperature over that on deck is noticed with oil fuel, whereas with coal under natural draught the stoke-hole temperature under similar external conditions is from 55° to 60° C. At the same time in the former case the stokers scarcely require to exert themselves at all, while in the latter, they come on deck every few minutes for a breath of air. Oil fuel would thus be very advantageous for steamers continually in the tropics.

Greater command of the machinery.

11) g. **Greater command of the machinery** is afforded by the possibility of urging, keeping back, or extinguishing the fires at any time. With coals, if it is desired to suddenly check the evaporation and at the same time avoid blowing-off at the safety-valves, the furnace and smoke-box doors must be opened and cold air allowed to pass through the boilers which is of course very injurious; and on the other hand, a considerable time elapses before a fire thus deadened can be restored to active combustion. Either object can be very rapidly achieved with liquid fuel by simply opening or closing the steam and oil cocks without doing the boiler the least harm.

Increased life of the boilers.

12) h. **The increased life of the boilers** is chiefly due to the rare intervals at which the fire-doors are opened, thus preventing the access of cold air to the internal parts of the boiler with its bad consequences. A less important consideration is that the oil-gases contain no sulphur**), so that the furnace plates are not attacked as they are with coal, the sulphur in which when combined with oxygen to sulphurous acid has a destructive effect on iron.

Simplicity of measurement.

13) i. **The simplicity and exactness of measurement** to which oil lends itself, as well at shipment as in use, would do away with the complaints of short weight which are not so rare in purchasing coals, and on the other hand would enable a better check to be kept upon the consumption of fuel at contractors' trial trips.

Enumeration of the disadvantages.

14) **II. The objections to liquid fuel,** which completely balance its advantages, although some of the latter are indeed brilliant, may be stated as follows,

*) Journal of the united service institution. May 1885. P. 697.

**) Verhandlungen des Vereins zur Beförderung des Gewerbfleisses. 1887. P. 546.

a) the necessity of establishing oil-holders and pipe-lines,
b) the great noise emitted by the jets,
c) the inflammability of the oil,
d) the small quantity of oil available, and
e) the high price of the oil.

15) a. **The establishment of oil-holders and pipe-lines**, instead of the existing coaling-stations, is necessary in order to take advantage of the before-mentioned possibility of rapid shipment. These oil-stations would not however require to be so numerous as coaling-stations, as steamers using oil fuel can travel further on the same weight than those using coal. The establishment and maintenance of such stations with efficient iron oil-holders, pumping arrangements for filling them and branch connections as numerous as possible for emptying, so as to supply a number of ships simultaneously, would not be much more costly than coaling-stations with well-kept-up sheds having the necessary spouts, stages, &c. The staff required for an oil-station is, on the other hand, extremely small compared with that of a coaling-station, as the distribution of the oil is effected entirely by mechanical appliances, whereas in the case of coals it is almost exclusively done by hand labour. However, if the military advantages of oil fuel are considered important enough, the expense of oil-stations will certainly not debar its adoption. Establishment of oil-stations.

16) b. **The very loud noise caused by the jets** is for passenger steamers a grave objection, but for torpedo-boats a fatal one. This noise, which is equally loud whether the jets are worked with steam or compressed air, and can only be diminished by keeping down the working pressure, is the chief obstacle to the use of oil fuel, particularly from a strategic point of view. Not until the ordinary pressure of 1.3 to 1.5 atmos. of the steam or air for the jets, has been allowed to sink to about 0.5 atmos., does the noise from the furnaces cease to drown that of the engines and pumps, but the speed which can then be kept up is only very moderate. Noise of the jets.

17) c. **The inflammability of the oil** may lead to a conflagration if the ship is hit. Against this it may be urged that the oil lends itself conveniently to stowage below the water-line where it is comparatively protected; and besides, it still requires to be ascertained by experiment whether the oil really takes fire on the holder being pierced by a fragment of a shell. The protection afforded to the boilers by coal bunkers placed above water must of course be abandoned; but this sacrifice is not very severe, as the thin bunker bulkheads and only moderate Inflammability.

thickness of coal can hardly be seriously taken into account against the destructive effect of modern shell fire.

Use of viscous oil.

18) It may here be mentioned that Admiral Selwyn proposes to employ only very heavy oil of 1.050 to 1.060 sp. g. — heavier than sea-water (1.026 sp. g.). Its advantages are evident. The thick oil is less easily ignited, is deprived of all volatile gases, burns more economically than limpid oil, and will not all escape into the sea if a shot should enter the bunker just below the water-line, as the light oils would. The heavy oil remaining in the bunker could be drawn off at the bottom and the bunker allowed to run up with sea water, increasing if anything, the vessels buoyancy, as the water is lighter than the oil. But when a coal-bunker is pierced it must be immediately shut off and the water entering it causes a certain loss of buoyancy.

Small available quantity of oil.

19) d. **The smallness of the quantity** of oil available for fuel is clearly evinced when we consider that the total annual production of crude oil in the world only amounts to about 6 million tons of petroleum, $^3/_4$ million tons of tar and about $^1/_4$ million tons of shale-oil, or together 7 million tons. It is indispensable for the industries of the present time, being required for the manufacture of illuminating and lubricating oils, benzol, paraffin, &c.; only the low-priced residue, amounting, at the outside, to 20 $^0/_0$ of the crude weight, can be spared for the purposes of fuel, which covers no more than about one-fifth of the requirements of the steam navigation of to-day. Whereas 12 million tons of coals suffice for this purpose or only 3 $^0/_0$ of the 400 million tons annually produced by the collieries of the world.

High expenses.

20) e. **The great expense** of oil-fuel is the rock on which all attempts have hitherto split, to introduce it on a general scale into the mercantile marine, in spite of its really splendid advantages. At the present prices of coal and petroleum-refuse, the latter is nearly three times as expensive fuel as the former, even taking into account the reduced stokehole staff and cheapness of shipment with oil.

Prospects of oil fuel.

21) Until new deposits of petroleum are discovered, or carriage is so far cheapened as to bring the price of petroleum refuse in western Europe to within twice that of the best coals, it will be impossible for oil fuel to seriously compete with coal. It appears however very doubtful whether even then such competition could be sustained for any length of time, experience in England and America having hitherto shewn that as soon as the demand for considerable quantities of oil, consequent upon starting a steamer fired with it, was felt, the price immediately rose so seriously that the owners could not make it

pay and were obliged to return to coals. But for warships, where expense is of less importance than in the mercantile marine, the only question is whether the much-boasted high strategic value of oil-fuel will be confirmed by the experiments still in progress in England, Russia, and France. Should this prove to be the case, as seems still uncertain, judging from recent trials of a Russian torpedo-boat by PASCHININ*) and an English one by DOXFORD & SONS**), it is probable that the introduction of oil-fuel for torpedo-boats in these countries, and perhaps in the United States, will still only be very gradual. It may however attain a certain importance in submarine boats on DE LAVAL's system, supposing his experiments to be succesful; and finally it may possibly come into use for steam yachts, a start in this direction having lately been made in America, because in both these cases, expense is no object. *There is certainly no immediate prospect of its introduction on a considerable scale either for war or merchant ships, attractive as its advantages may be, except on the Caspian and the rivers of southern Russia.*

*) Morskoi Skornik. 1887. December. P. 39.
**) Engineering. 1890. I. P. 32.

Fifth Division.

Determination of the Power and Economy of Marine Engines.

§ 25.

Loss of Pressure at Admission.

Resistance in the pipes and cooling.

1) Steam issuing from the boiler at the pressure p_x suffers on its way to the cylinder a *loss of pressure z*, in consequence of friction in passing through the stopvalves, pipes, slides, and ports. Besides this there is a loss of temperature in the pipes caused by external cooling, accompanied by a certain amount of condensation, and therefore a diminution of the mass and volume of the steam. These losses by condensation are more closely discussed later on in the paragraphs on "Steam-pipes", whereas here only the loss of pressure to be expected is referred to.

Estimation of the loss of pressure.

2) Experiments of NASSE, EHRHARDT, and GUTERMUTH*) carried out on a range of steam-piping partly of cast and partly of wrought iron, have shewn that the loss of pressure

$$z = 0{,}0015\,\gamma\,\frac{l}{d}\,u^2 \text{ kg per } \square \text{ cm}, \quad . \; . \; . \; . \; . \quad (108)$$

assumed by H. FISCHER**) in calculating the most economical diameter for steam-pipes, agrees with actual results. In this formula

z is the diminution of pressure in atmos.,
γ the specific weight of the steam (see Table, p. 28),
l the length of the pipes in m,
d the diameter of the pipes in m, and
u the velocity of the steam per sec. in m.

*) Zeitschrift des Vereines deutscher Ingenieure. 1887. P. 718.
**) DINGLER'S polytechnisches Journal. 1880. Vol. 236. P. 354.

3) As however the theoretical estimation of the diminution of pressure, according to GRASHOF *), always necessitates certain simplifying assumptions in order to produce results sufficiently free from complication, and as the above formula is correct only when z and u are not too large, it is more exact to estimate the diminution of pressure in marine engines from actual trial-trip observations. Inaccuracy of the formula.

4) On the trial trips of German warships, the loss of pressure z has shewn itself to vary pretty uniformly between 0.4 and 0.6 atmos., within the limits of usual steam velocities, as well for high as low pressures, and according to the length of the pipes and the number of obstacles (stopvalves, water-separators, &c.), all being well cleaded with felt and canvas. As a suitable average value for marine engines we may therefore take Loss of pressure as found by experience.

$$z = 0.5 \text{ atmos.} \quad \ldots \ldots \ldots \ldots \quad (108^{a})$$

5) The *admission-pressure* p for the *HP* cylinder of a marine engine may consequently be expected in round numbers to be Admission pressure.

$$p = p_x - z = p_x - 0.5 \text{ atmos.} \quad \ldots \ldots \ldots \quad (109)$$

§ 26.

Throttling.

1) In order that a marine engine may develop its greatest economy, it is necessary that the range of piping from the boiler to the cylinder should contain no obstacles of any kind to the motion of the steam, so that the diminution of pressure may be very small, i. e. that the admission pressure may approach as nearly as possible to the boiler pressure. Any alteration in the number of revolutions, or in other words, of the power of the engine, ought properly to be brought about by an alteration of the cut-off alone. *Throttling* should only be permitted when manœuvring under steam and when "standing-by" in bad weather. Throttling is produced by contracting the free passage of the steam either by means of a throttle-valve or the stop-valve. Prohibition of throttling.

2) The experience of many years has proved that any engine works more economically with the stopvalves &c. opened full and a certain cut-off, than when producing the same power with a later cut-off but throttled. Recently this fact has been demonstrated by, among others, DONKIN and SALTER **), with Wastefulness of throttling.

*) F. GRASHOF. Theoretische Maschinenlehre. Vol. III. Brunswick 1888. P. 569.
**) Engineering 1886. II. P. 573.

their experimental engine already mentioned on p. 98. They found that a compound engine with 2.8 to 2.95 atmos. admission pressure, cutting off in the *HP* cylinder at $^{3}/_{16}$ to $^{7}/_{16}$ of the stroke, shewed 12 % more economy than when the steam was throttled down to 1 atmos. admission pressure and the cut-off in the *HP* cylinder was at $^{7}/_{8}$ of the stroke. Fig. 1, Pl. 8 shews a set of indicator cards taken at these trials, the full lines referring to the unthrottled and the dotted lines to the throttled condition.

Single case in which throttling is economical.

3) If however an engine is occasionally required to work for a short period with a weight of steam per stroke greater than the boiler is capable of generating continuously for a length of time, then for any cut-off earlier than half-stroke it is more economical to keep up the pressure in the boiler to its highest limit, by means of throttling, than to open out and work with a later cut-off and less pressure, as ROSENKRANZ*), in particular, has shewn most convincingly, — see Fig. 2. Pl. 8. In this, ABC is the opened-out diagram and DEF the throttled one, the areas being of course equal, as the powers are. It is evident that the pressure BG is greater than EG (the volumes being equal), therefore the *weight* of steam for the same *IHP*., is less when opened out, than when throttled.

Further injurious effect of throttling.

4) For an engine with a later cut-off than half-stroke, for instance, a compound engine carrying the steam the whole stroke in the *HP* cylinder, throttling is however, always accompanied with loss. Up to half stroke, it causes the pressure to fall and afterwards to rise again to nearly its initial height, as the piston speed is retarded in the second half of the stroke. The work in the cylinder is therefore less than it would be with the same weight of steam and constant pressure, that is with a lower admission pressure and no throttling.

Former views as to the utility of throttling.

5) The statement of SCHMIDT and BOCKHOLTZ**) to the effect that throttling is economical is, by the foregoing, only true to a very limited extent. The same applies to ZEUNER'S***) inference that the admission work of a condensing engine cutting off at $^{1}/_{4}$ stroke, is slightly more economical when the pressure is kept at 7 atmos. in the boiler and throttled down to 3.5 atmos. at admission, than when the steam is generated at the lower pressure and allowed to pass entirely without

*) P. H. ROSENKRANZ. Der Indicator und seine Anwendung. Edit. IV. Berlin 1885. P. 93.

**) Mittheilungen des Architecten- und Ingenieur-Vereins in Böhmen. 1874. Number 1.

***) Der Civilingenieur. Leipzig 1875. P. 1.

obstruction into the cylinder. According to GRASHOF*), this advantage would probably quite disappear if the equal total works were properly compared together, instead of the equal admission works.

6) If it be true that a slight gain may be expected from throttling, this must be due to the high kinetic energy, acquired by the steam in passing rapidly through the contracted valve-opening, becoming afterwards converted into *heat* in consequence of the formation of eddies behind the obstruction. This heat serves to evaporate the watery particles carried over or, in other words, to dry the steam. The makers of water-tube boilers, in particular, have of late attempted to turn this circumstance to account (see § 55, 27). BELLEVILLE**) for instance, works his recent water-tube boilers at a pressure 5 to 6 atmos. higher than that of the admission in the *IP* cylinder, throttling down the steam by means of a special throttle valve in the steam pipe; the avowed intention being to dry the somewhat wet steam always produced in his boilers, even when quite moderately worked. **Advantage of throttling.**

7) NIMAX***) goes still further in this direction, by asserting not only that throttling the steam for the purpose of drying it, is indispensable to economy, but also attempting by throttling to a sufficient extent, to obtain the advantage due (§ 16, 26, p. 91) to jacketing the cylinder with steam of a higher pressure than the working steam. NIMAX accounts for this by the empirical statement that no more fuel is required in producing a certain weight of high than of low pressure steam. He forgets however that the weights and therefore the cost of ordinary cylindrical marine boilers increase quite disproportionately as the working pressure is raised, because of the heavier scantlings required. It would be a great mistake to have a triple with 12 atmos. admission pressure and work the boilers at 18 atmos. for the sake of effective throttling. NIMAX'S proposal may however be worth further consideration as applied to water-tube boilers, because their working pressure may be taken tolerably high without any particular increase of weight and cost. But for the ordinary working of marine boilers at present, the old rule, "*no throttling*", is still in force. **Advisability of throttling.**

*) F. GRASHOF. Theoretische Maschinenlehre. Hamburg 1888. Vol. III. P. 574.

**) C. BUSLEY. Die Entwickelung der Schiffsmaschinen. Edit. II. Berlin 1870. P. 164.

***) Zeitschrift des Vereins deutscher Ingenieure. 1884. P. 286.

§ 27.

Ratio of cut-off.

Ratio of cut-off.

1) If we call that portion of the stroke described by the piston during admission h, then for single-expansion engines, the ratio of h to the whole stroke H, or

$$\frac{h}{H} = \varepsilon, \textit{ the ratio of cut-off,}$$

$$\text{and } \frac{H}{h} = \frac{1}{\varepsilon}, \textit{ the ratio of expansion.}$$

Total ratio of cut-off.

2) In multiple-expansion engines ε is the ratio of cut-off in the HP. cylinder, while

$$\varepsilon_g = \varepsilon \frac{v}{V} \textit{ is the total ratio of cut-off,}$$

$$\text{and } \frac{1}{\varepsilon_g} = \frac{1}{\varepsilon}\frac{V}{v} \textit{ is the total ratio of expansion,}$$

where v is the volume of the HP and V that of the LP cylinder or cylinders.

Actual ratio of cut-off.

3) As was shewn in § 18, 51 (p. 141), the cylinder of a single-expansion engine of the diameter D receives at each stroke not only the volume of steam corresponding to h, the portion of the stroke described up to the moment of cut-off $\varepsilon h \frac{\pi}{4} D^2$, but also the volume of steam required to bring up the pressure in the clearance from the compression pressure to the admission pressure. This volume, represented in Fig. 10, Pl. 2 by the distance SC may be regarded as the m^{th} part of the volume swept, and expressed as $m H \frac{\pi}{4} D^2$. The consumption of steam of a single-expansion engine thus corresponds to

$$\varepsilon' = \varepsilon + m, \textit{ the actual ratio of cut-off}$$

$$\text{and } \frac{1}{\varepsilon'} = \frac{1}{\varepsilon + m}, \textit{ the actual ratio of expansion.}$$

For a multiple-expansion engine we should then have (see 2)

$$\varepsilon'_g = (\varepsilon + m) \frac{v}{V}, \textit{ the actual total ratio of cut-off}$$

$$\text{and } \frac{1}{\varepsilon'_g} = \frac{1}{\varepsilon + m} \frac{V}{v}, \textit{ the actual total ratio of expansion.}$$

Most economical ratio of cut-off.

4) **The most economical ratio of cut-off** for any steam engine is that at which the water used per horse-power per hour is a minimum.

Determination of the best ratio of cut-off.

5) The best ratio of cut-off ε_v must be specially fixed for every engine according to the admission pressure p and back-

pressure a by calculating (§ 31) the water used for various cut-offs and choosing that one which requires the least water.

6) The most profitable total ratios of cut-off ε_{g_v} for the various types of multiple-expansion engines are about the following, viz. Best total ratios of cut-off as found by experience.
   0.060 for the latest quadruple-expansion engines of 14 to 15 atmos. working pressure,
   0.065 for recent triple-expansion engines of 10 to 12 atmos. working pressure,
   0.085 for earlier triple-expansion engines of 8 to 10 atmos. working pressure,
   0.100 for recent compound engines of 6 to 7 atmos. working pressure and high piston-speed,
   0.150 for older compound engines of 4 to 5 atmos. working pressure and lower piston-speed,
   0.250 for medium-pressure compound engines of 3 atmos. working pressure with surface condensers, jackets, and superheaters.

7) The most advantageous ratio of cut-off ε_v for single-expansion engines with 2 atmos. working pressure is about Best ratios of cut-off for single expansion engines as found by experience.
   0.35 for engines having surface-condensers, jackets, and superheaters,
   0.40 for engines having jet-condensers, and neither jackets nor superheaters,
   0.60 for old engines, in other respects as the last, but with less than 2 atmos. working pressure.

8) In trial-trips, held with the object of ascertaining the highest speed of the ship, the multiple-expansion engines of modern warships must always be worked at a greater total ratio of cut-off than the most economical one, whereas formerly with the old single-expansion engines it was only during forced trials that the most favourable ratio of cut-off could be kept up at all. At a lower speed of the ship the latter always worked with an earlier cut-off than the most economical one. With the triple-expansion engines usual in modern cruisers and torpedo-boats, the object is to keep as an average to that particular speed which corresponds to the most economical cut-off of their engines, — or a disconnected portion of them. The triple-expansion engines of ordinary mail and cargo boats are designed to attain their regular average speed with the the most economical cut-off. Only the most recent "Greyhounds" work with later cut-offs. Application of the most favourable ratio of cut-off in practice.

§ 28.

Back pressure.

1) The back pressure a (see § 9, 7, p. 25) which opposes the motion of the piston in the cylinder of a single-expansion Amount of back-pressure according to experience.

engine or the L. P. cylinder of a multiple-expansion engine, arises, in non-condensing engines, from the pressure of the atmosphere, and in non-condensing engines, from the condenser-pressure, augmented in both cases by the friction which the exhaust steam has to overcome in the ports and pipes. As a general rule we may assume

for non-condensing engines . . . . $a = 1.1$ atmos.
" jet-condensing engines . . . . $a = 0.2$ "
" surface-condensing engines . . . . $a = 0.15$ "

Lowest back-pressure.

2) In marine engines, it is advisable, on account of the unlimited supply of cooling water, to keep the back pressure a as low, or in other words, the vacuum as high, as possible. It is true that the weight of cooling water increases rapidly with the rising vacuum, and therefore also the work of the pumps; but as the water runs into them of itself and only requires to be lifted about 2 to 3 m, the gain in indicated work due to the lower back pressure is always greater than the loss of useful work due to the increased duty of the pumps. — With land-engines on the other hand, it is not always advantageous to work with the smallest attainable back pressure, particularly when the supply of cooling water is limited, as WEISS*) has recently convincingly shewn.

Terminal pressure.

3) The terminal pressure, shewn at point 3 of the diagram Fig. 6, Pl. 2 is not to be confounded with the back pressure a exerted at points 4 and 5 of this diagram. Even if, according to RADINGER,**) it is under certain circumstances profitable to allow the steam to expand down to the back pressure a in single-expansion engines, which in the above diagram would cause the points 3 and 4 to coincide, this is not advantageous in multiple-expansion engines. The late Mr. WYLLIE,***) in his triples, which were patterns of careful design, only carried his terminal pressure down to 0.7 atmos. absolute, while keeping the back pressure as low as possible. The object of this was to avoid too large a L P. cylinder, the heavy rods of which would interfere with the smooth working of the engine.

§ 29.

Mean Pressure.

Work during admission.

1) I. **Theoretical mean pressure** p_m. In *the period of admission* the steam moves the piston h metres under an approximately constant

*) Zeitschrift des Vereines deutscher Ingenieure 1889. P. 772.

**) J. F. RADINGER. Ueber Dampfmaschinen mit hoher Kolbengeschwindigkeit. II. Edit. Vienna 1872. P. 56.

***) Proceedings of the institution of mechanical engineers. 1886. P. 475.

pressure p (see Fig. 3, Pl. 8). Assuming the piston area to be 1 □cm, the *work during admission* is

$$p\,h \quad \text{mk*)}$$

2) During the expansion Mariotte's law is in force by § 18, 29, p. 131, so that **Law of expansion.**

$$p\,v = p_1\,v_1.$$

As the initial volume $v = 100 \times h \times 1$ and the volume at any portion h_1 metres of the stroke during expansion is

$$v_1 = 100 \times h \times 1 \text{ cb. cm},$$

$p\,h$ must also be $= p_1\,h_1$, and the pressure at the stroke h_1 must be

$$p_1 = \frac{p\,h}{h_1}.$$

3) For an infinitely small forward movement of the piston $\varDelta$, this pressure may be regarded as constant, and the *element of work* then performed is **Element of work.**

$$p_1\,\varDelta = p\,h\,\frac{\varDelta}{h_1}.$$

If in the series

$$e^x = 1 + x + \frac{x^2}{1 \times 2} + \frac{x^3}{1 \times 2 \times 3} + \ldots\ldots$$

x is regarded as infinitely small, all the terms containing powers of x may be neglected, and we have, without perceptible error

$$e^x = 1 + x$$

or

$$x = log\,nat\,(1 + x).$$

In the present case $\frac{\varDelta}{h_1}$ is an infinitesimal quantity, for which, analogously to the above, we may write

$$\frac{\varDelta}{h_1} = log\,nat\left(1 + \frac{\varDelta}{h_1}\right)$$

so that the element of work may be expressed as

$$p_1\,\varDelta = p\,h\left[log\,nat\left(1 + \frac{\varDelta}{h_1}\right)\right]$$

$$p_1\,\varDelta = p\,h\left[log\,nat\left(\frac{h_1 + \varDelta}{h_1}\right)\right]$$

$$p_1\,\varDelta = p\,h\left[log\,nat\,(h_1 + \varDelta) - log\,nat\,h_1\right].$$

4) The work done by the piston during the further course of the expansion is the sum of all the elements of work. **Expansion work up to any point.**

This is obtained by successively substituting in the above equation instead of $h_1 + \varDelta_1$ the values $h_1 + 2\,\varDelta$, $h_1 + 3\,\varDelta$, &c., up to $h_1 + n\,\varDelta$ (n being regarded as infinitely great); and for

*) *mk* = kilogrammetres.

h, the values $h_1 + \Delta$, $h_1 + 2\Delta$, &c., up to $h + (n-1)\Delta$. Carrying out this summation, we get

$$\Sigma(p_1 \Delta) = p\,h \begin{cases} \log nat\,[h_1 + \Delta] - \log nat\, h_1 \\ + \log nat\,[h_1 + 2\Delta] - \log nat\,[h_1 + \Delta] \\ + \log nat\,[h_1 + 3\Delta] - \log nat\,[h_1 + 2\Delta] \\ \quad \vdots \qquad\qquad\qquad\qquad \vdots \\ + \log nat\,[h_1 + (n-2)\Delta] - \log nat\,[h_1 + (n-3)\Delta] \\ + \log nat\,[h_1 + (n-1)\Delta] - \log nat\,[h_1 + (-2)\Delta] \\ + \log nat\,[h_1 + n\Delta] - \log nat\,[h_1 + (n-1)\Delta]. \end{cases}$$

In this equation, one term in each line goes out with one term in the next line, and it ultimately assumes the form

$$\Sigma(p_1 \Delta) = p\,h\,[\log nat\,(h_1 + n\Delta) - \log nat\, h_1].$$

Total work of expansion.

5) If the summation is begun at h, corresponding to the cut-off point, instead of at an arbitrarily chosen position h_1, then the whole work done during expansion is given by the following expression

$$p\,h\,[\log nat\,(h + n\Delta) - \log nat\, h].$$

but $$h + n\Delta = H$$

$$p\,h\,[\log nat\, H - \log nat\, h] = p\,h \log nat \frac{H}{h}$$

or, referring to the definitions in § 27, 1, p.

$$p\,h \log nat \frac{H}{h} = p\,h \log nat \frac{1}{\varepsilon} \text{ mk.}$$

Total work during one stroke.

6) The above result may be much more shortly arrived at by help of the integral calculus. If $d\,h_1$ is an infinitesimal portion of the stroke, the element of work becomes

$$p_1\, d\,h_1 = p\,h \frac{d\,h_1}{h_1}.$$

and the whole work during expansion is

$$\int_h^H p\,h \frac{d\,h_1}{h_1} = p\,h\,(\log nat\, H - \log nat\, h) = p\,h \log nat \frac{H}{h} = p\,h \log nat \frac{1}{\varepsilon} \text{ mk.}$$

Separating the work done during admission (see 1), we get the following expression for the total work per stroke

$$p\,h + p\,h \log nat \frac{1}{\varepsilon} = p\,h\left(1 + \log nat \frac{1}{\varepsilon}\right) = p\,H\,\varepsilon\left(1 + \log nat \frac{1}{\varepsilon}\right) \text{ mk.}$$

Theoretical mean pressure p_m.

7) That pressure per $\square$cm area, which, being multiplied by the whole stroke H, produces the above work per stroke, is called *the theoretical mean pressure* p_m

$$p_m H = p\,H\,\varepsilon\left(1 + \log nat \frac{1}{\varepsilon}\right) \text{ mk}$$

$$p_m = p\,\varepsilon\left(1 + \log nat \frac{1}{\varepsilon}\right) \text{ atmos.} \quad \ldots\ldots \quad (110)$$

8) **II. Reduced mean pressure p_{m_r}.** As appears from what is said on the subject of combining the diagrams of multiple-expansion engines in § 18, V, it is theoretically indifferent whether the steam expands from an initial pressure p with a total ratio of cut-off ε_g *only* in the LP cylinder of a multiple-expansion engine, or in the several cylinders of such an engine, provided it starts with the same cut-off ratio ε and the same pressure in the HP cylinder. In the former case the mean pressure of the multiple-expansion engine would be expressed, according to Eq. 110 as

General formula for the reduced mean pressure p_{m_r}.

$$p_{m_r} = p\,\varepsilon_g \left(1 + log\ nat \frac{1}{\varepsilon_g}\right) \dots\dots\dots\dots (111)$$

In the latter case, the steam, while acting in the H. P. and intermediate cylinders, would produce the same work in the LP cylinder as it would if its mean pressure were reduced in the inverse ratio of the cylinder-volumes or, when the strokes are equal, of the piston-areas.

9) *In a 2-cylinder compound engine*, the reduced mean pressure must therefore be

Reduced mean pressure in 2-cylinder compound engines.

$$p_{m_r} = \frac{v}{V} p_{m_h} + p_{m_n} \dots\dots\dots\dots (112)$$

where p_{m_h} and p_{m_n} are the mean pressures in the HP and LP cylinders respectively. Further

$$p_{m_h} = p\,\varepsilon \left(1 + log\ nat \frac{1}{\varepsilon}\right) \dots\dots\dots\dots (113)$$

and

$$p_{m_n} = p_n\,\varepsilon_n \left(1 + log\ nat \frac{1}{\varepsilon_n}\right) \dots\dots\dots\dots (113^a)$$

if p_n is the initial pressure and ε_n the ratio of cut-off in the LP cylinder. Consequently we also have

$$p_{m_r} = \frac{v}{V} p\,\varepsilon \left(1 + log\ nat \frac{1}{\varepsilon}\right) + p_n\,\varepsilon_n \left(1 + log\ nat \frac{1}{\varepsilon_n}\right) \dots (114)$$

10) *In a 3-cylinder compound engine* the reduced mean pressure is

Reduced mean pressure in 3-cylinder compound engines.

$$p_{m_r} = \frac{v}{V} p_{m_h} + \frac{1}{2}\left(p'_{m_n} + p''_{m_n}\right) \dots\dots\dots\dots (115)$$

where V is the combined volume of the two LP cylinders and p'_{m_n} and p''_{m_n} are the respective mean pressures in these cylinders, which are calculated from Eq. 113a after the proper substitutions.

11) *For a triple-expansion engine*, the reduced mean pressure becomes

Reduced mean pressure in multiple-expansion engines.

$$p_{m_r} = \frac{V_h}{V_n} p_{m_h} + \frac{V_m}{V_n} p_{m_m} + p_{m_n} \dots\dots\dots\dots (116)$$

in which

V_h = the volume, and p_{m_h} the mean pressure in the HP cylinder

V_m = the volume, and p_{m_m} the mean pressure in the MP cylinder
V_n = " " " p_{m_n} " " " " " LP "

p_{m_h} is calculated from Eq. 113, while p_{m_m} and p_{m_n} are obtained by substituting the proper values in Eq. 113a. — If the triple has two LP cylinders, V_n must be put = to their combined volume, and $\frac{1}{2}\left(p'_{m_n} + p''_{m_n}\right)$ inserted instead of p_{m_n}. For a quadruple p_{m_r} is determined analogously.

Reduced diameter of the LP cylinders.

12) With multiple-expansion engines having *two* or *three* LP cylinders, it is sometimes convenient to regard the latter as replaced by *one* LP cylinder of a piston area equal to the combined areas of the actual LP cylinders.

The diameter of this cylinder D_r would then be

$$\frac{\pi}{4} D_r^2 = 2\frac{\pi}{4} D^2 \quad \text{or} \quad = 3\frac{\pi}{4} D^2$$

$$D_r = \sqrt{2}\, D \quad \text{or} \quad = \sqrt{3}\, D \quad \dots\dots\dots \quad (117)$$

and this imaginary diameter is called the *reduced diameter* of the LP cylinders.

Application of the table.

13) **III. Actual mean pressure p_0.** If we calculate from Eq. 110 or 111, for different values of ε, or, what is the same in this case, of ε_g, the factor

$$\zeta = \varepsilon\left(1 + \log nat \frac{1}{\varepsilon}\right) = \varepsilon_g\left(1 + \log nat \frac{1}{\varepsilon_g}\right) \quad \dots\dots \quad (118)$$

the theoretical reduced mean pressure is

$$p_m = \zeta p \text{ or } p_{m_r} = \zeta p \text{ atmos.} \quad \dots\dots\dots \quad (119)$$

and the mean pressure can be easily found from the following table by multiplying the initial pressure p by the factor ζ, corresponding to the ratio of cut-off ε or ε_g.

Mean ordinate of theoretical diagrams.

14) The values of p_m found by help of the table correspond to the mean ordinate of the theoretical diagram § 18, 14 to 30, p. 125 (Fig. 6, Pl. 2); and the values of p_{m_r} refer to the mean ordinate of the theoretical diagram described by COWPER and RANKINE'S method (§ 18, 39 to 46, p. 136 and Figs. 8 and 9, Pl. 2), the clearance $ONCA$ in these diagrams being neglected for the reasons explained in § 18, 44, g. 138.

Actual mean pressure.

15) In single-expansion engines, *the actual mean pressure p_0* differs as considerably from *the theoretical mean pressure p_m*, as the form of the diagram Fig. 6, Pl. 2 (after deducting the area $ONCA$) does from that of the enclosed indicator diagram. In other words, *the reduced mean pressure p_{m_r}* must be as many times greater than the *reduced actual mean pressure p_{0_r}* of a multiple-expansion engine, as the shaded area of the enclosing diagram Fig. 9, Pl. 2 is greater than the unshaded area of the indicator diagram.

Table for calculating the theoretical and reduced mean pressure p_m and p_{m_r}.

cut-off ratio ε	Factor ζ	Differences	cut-off ratio ε	Factor ζ	Differences	cut-off ratio ε	Factor ζ	Differences	cut-off ratio ε	Factor ζ	Differences
1	2	3	1	2	3	1	2	3	1	2	3
0,01	0,0561	561	0,26	0,6102	136	0,51	0,8534	68	0,76	0,9686	28
0,02	0,0982	421	0,27	0,6235	133	0,52	0,8600	66	0,77	0,9713	27
0,03	0,1352	370	0,28	0,6365	130	0,53	0,8665	65	0,78	0,9738	25
0,04	0,1688	336	0,29	0,6490	125	0,54	0,8727	62	0,79	0,9762	24
0,05	0,1998	310	0,30	0,6612	122	0,55	0,8788	61	0,80	0,9785	23
0,06	0,2288	290	0,31	0,6731	119	0,56	0,8847	59	0,81	0,9807	22
0,07	0,2561	273	0,32	0,6846	115	0,57	0,8904	57	0,82	0,9827	20
0,08	0,2821	260	0,33	0,6959	113	0,58	0,8959	55	0,83	0,9847	20
0,09	0,3067	246	0,34	0,7068	109	0,59	0,9013	54	0,84	0,9865	18
0,10	0,3303	236	0,35	0,7174	106	0,60	0,9065	52	0,85	0,9881	16
0,11	0,3528	225	0,36	0,7278	104	0,61	0,9115	50	0,86	0,9897	16
0,12	0,3744	216	0,37	0,7379	101	0,62	0,9164	49	0,87	0,9912	15
0,13	0,3952	208	0,38	0,7477	98	0,63	0,9211	47	0,88	0,9925	13
0,14	0,4153	201	0,39	0,7572	95	0,64	0,9256	45	0,89	0,9937	12
0,15	0,4346	193	0,40	0,7665	93	0,65	0,9300	44	0,90	0,9948	11
0,16	0,4532	186	0,41	0,7756	91	0,66	0,9342	42	0,91	0,9958	10
0,17	0,4712	180	0,42	0,7844	88	0,67	0,9383	41	0,92	0,9967	9
0,18	0,4887	175	0,43	0,7929	85	0,68	0,9422	39	0,93	0,9975	8
0,19	0,5055	168	0,44	0,8012	83	0,69	0,9460	38	0,94	0,9982	7
0,20	0,5219	164	0,45	0,8093	81	0,70	0,9497	37	0,95	0,9987	5
0,21	0,5377	158	0,46	0,8172	79	0,71	0,9532	35	0,96	0,9992	5
0,22	0,5531	154	0,47	0,8249	77	0,72	0,9565	33	0,97	0,9995	3
0,23	0,5680	149	0,48	0,8323	74	0,73	0,9597	32	0,98	0,9998	3
0,24	0,5825	145	0,49	0,8395	72	0,74	0,9628	31	0,99	0,9999	1
0,25	0,5966	141	0,50	0,8466	71	0,75	0,9658	30	1,00	1,0000	1

16) As single-expansion marine engines are now almost obsolete, the term *actual mean pressure* p_0 will, in the following pages, be used for the sake of simplicity with reference to that of multiple-expansion engines (properly *reduced* p_{0_r}), where no ambiguity can arise. The formulæ then become applicable both to single and multiple-expansion engines. **Simplified terms.**

17) *Discrepancies between the actual and theoretical diagrams* are due to **Discrepancy between theoretical and actual diagrams.**

a) the expansion of the steam being only approximately represented by Mariotte's law (§ 18, 29, p. 131);
b) the steam not being expanded, as assumed in the theoretical diagram, down to an absolute vacuum, but allowed to enter the condenser at a certain *back-pressure a* (see § 28, 1, p. 239);
c) the admission, release, compression, &c., varying according to the peculiarities of the valve-gear used, as described later on;
d) the losses of pressure always occurring during the passage of the steam between the cylinders of multiple-expansion engines.

Coefficient of fulness β.

18) Taking all these circumstances into account, and using, instead of the somewhat unsteady admission pressure p, the more constant boiler pressure p_x, we get the *actual mean pressure* p_0, if, instead of the theoretical mean pressure

$$p_m = \zeta p$$

we write

$$p_0 = \beta \zeta p_x$$

or

$$p_0 = \beta p_x \varepsilon \left(1 + \log nat \frac{1}{\varepsilon}\right) \quad \ldots\ldots\ldots\ldots (120)$$

in which equation β is the *coefficient of fulness* (§ 18), depending upon the type of engine and the arrangement of valve-gear. But it must be expressly noted that β is not the *absolute fulness*, expressed by Eq. 91, p. 133, that is, that it does not refer to the weight of feed water used, but to the steam shewn by the card at the beginning of the expansion. The coefficient β therefore expresses the ratio of the area of the combined indicator diagram to that of the enclosing diagram formed by a Mariotte's curve laid through the HP cut-off point, — the clearance being neglected, as remarked in 14).

Initial pressure p.

19) The initial pressure p may be approximately calculated by Eq. 120 from the mean pressure p_0 and the cut-off ratio ε (or total cut-off ratio ε_g)

$$\left.\begin{aligned} p = p_x - 0.5 &= \frac{p_0}{\beta \varepsilon \left(1 + \log nat\right) \frac{1}{\varepsilon}} - 0.5 \\ &= \frac{p_0}{\beta \zeta} - 0.5 \quad \text{atmos} \end{aligned}\right\} \quad (121)$$

Multiplying β with the factor ζ from the table on p. 245 and dividing the mean pressure p_0 by this product, we get the boiler pressure p_x and, deducting the loss of pressure, the approximate initial pressure p.

20) The cut-off ratio ε, or the total cut-off ratio ε_g can also be obtained by means of the table on p. 245 from the boiler pressure p_x and the mean pressure p_0, for as Cut-off ratio.

$$\zeta = \frac{p_0}{\beta p_x} \quad \ldots \ldots \ldots \ldots \ldots \quad (122)$$

we need only form this quotient and find from the table the cut-off ratio ε corresponding to this value of ζ. The differences for ζ are given in the table for the purpose of interpolation.

21) **IV. General values of β.** *For single-expansion condensing engines* calculations made with this particular object have given the following values of the coefficient β. Coefficient β for single-expansion engines.

a) From the cards taken on the trial trips of the older German war-ships, it was found that

$\beta = 0.56$ to 0.60 for engines of 2 atmos. working pressure with cam valve-motion,

$\beta = 0.56$ to 0.58 for the same engines with link-motion,

$\beta = 0.60$ to 0.70 for engines of 4 to 5.5 atmos. working pressure with link-motion.

The lower values refer to earlier, the higher ones to later cut-offs, and all the above values are in general rather under the truth, because in most cases the boilers were so small that the engines never indicated the power for which they were designed.

b) In EMERY'S trials of American marine engines, see tables on p. p. 85 and 86

$\beta = 0.60$ to 0.70 for unjacketed engines,

$\beta = 0.70$ to 0.80 for jacketed engines.

These values, like those more recently obtained by him, are rather too high, because the trials were not carried out with the requisite scientific exactness.

c) In the French Navy the following results were reported by the Inspecteur général du génie maritime*)

$\beta = 0.56$ to 0.84, mean $\beta = 0.65$ for 204 observations of screw-engines,

$\beta = 0.68$ for 31 observations of paddle-engines.

On the basis of these figures, the *average for single-expansion engines* may be put at

$$\beta = 0.60 \text{ to } 0.66.$$

The value $\beta = 0.60$ increases with the boiler pressure, the cut-off ratio, and the completeness of the jacketing.

22) *For compound engines* the following data exist, viz. β for compound-engines.

*) A. LEDIEU. Nouvelle théorie élémentaire des machines à feu etc. Paris 1882. P. 627.

I. For Woolfe engines;

a) according to the above French report,

$\beta = 0.63$ to 0.76 or 0.68 mean, from 78 observations of marine engines working at 1.75 to 4 atmos. boiler pressure;

b) in EMERY'S experiments (Table, p. 87) with a working pressure of 5.5 to 5.7 atmos., the mean value was

$\beta = 0.58$ without jacketing,

$\beta = 0,75$ with jacketing;

c) on trial-trips of German war-ships,

$\beta = 0.65$ to 0.68 at working pressures of 5 to 7 atmos.

As an average value for Woolf engines, we may therefore take

$$\beta = 0.65 \text{ to } 0.68$$

II. For two and three-cylinder compound engines,

a) on trial-trips of recent German war-ships

$\beta = 0.66$ to 0.72 for screw engines with 5 to 7 atmos. working pressure and jackets;

b) on EMERY'S trials (Table, p. 88)

$\beta = 0.77$ on an average, with 2.5 to 5 atmos. working pressure and jackets;

c) according to the French report

$\beta = 0.68$ for compound engines of medium working pressure.

As a good average for modern compound engines of careful design we may thus take

$$\beta = 0.66 \text{ to } 0.70.$$

β for triple-expansion-engines.

23) *For triples*, especially some of the best of the early ones,

a) OTTO H. MÜLLER jun.*) found

$\beta = 0.64$ to 0.67 for screw engines of 8 to 11 atmos. working pressure;

b) for WYLLIE'S **) well-known engines, the value varied between

$\beta = 0.60$ and 0.70 for screw engines working at 9.5 to 10.5 atmos.;

c) for engines belonging to the German Navy tried hitherto,

β averages between 0.64 and 0.68 for screw engines of 10 to 13 atmos. working pressure.

The average value for good triples may therefore be taken at

$$\beta = 0.68 \text{ to } 0.70.$$

In some very successful engines a still higher value than 0.70 has been reached.

β for quadruples.

24) *For quadruple-expansion engines*, there are as yet but very few available data, and the best known engines of this type cannot

*) Zeitschrift des Vereines deutscher Ingenieure. 1887. P. 446.

**) Proceedings of the institution of mechanical engineers. 1886. P. 473.

be regarded as examples worthy of imitation. From an examination of the indicator cards of four of these,*) the author obtained an average value of $\beta = 0.59$. Three of the sets of cards are given, viz.

"County of York". Fig. 9, Pl. 2, where $\beta = 0.59$,
"Rionnag-na-Mara" „ 4, „ 3, „ $\beta = 0.59$,
and "Suez" .... „ 1, „ 4, „ $\beta = 0.60$.

The working pressure of all these engines is too low for quadruple expansion.

If the pressure were raised to 14 or 15 atmos., as has recently been done, the fulness of the diagrams would come out better. With carefully designed cylinders and valve-gear, almost the same coefficient β must be got as in good triples. As a proof of this assertion we may adduce ADAMSON'S **) stationary quadruple engine of 1875, the combined diagram of which shews a fulness of 0.71.

If therefore, *in designing a quadruple engine*, we take

$$\beta = 0.66,$$

it is sufficiently certain that the corresponding mean pressure will be actually got.

25) In conclusion it may be mentioned that considerably higher values of β have been obtained with stationary engines than has as yet been the case with marine engines. A two-cylinder compound engine built by the Prager Maschinenbau Actien Gesellschaft in 1885 and tested by DOERFEL ***) gave $\beta = 0.76$, and as mentioned on p. 133, single-expansion Corliss engines are stated to have shewn as high a value for β as 0.85.

Greatest values of β hithertho reached.

§ 30.

Horse-power.

1) In countries where the metrical system is used, the work expended in raising 75 kg one metre high in one second is called *a horse-power*;

Metrical Horse-power.

$$1\ IP = 75 \text{ smk.}$$

2) In England, North America, and the other countries using English units, a horse-power means the work of raising 550 pounds avoirdupois one foot high in one second, so that

English Horse-power.

$$1\ IP = 550 \text{ foot pounds per second.}$$

*) C. BUSLEY. Die Entwickelung der Schiffsmaschine in den letzten Jahrzehnten. Edit. II. Berlin 1890. P. 62.

**) Quadruple engines. Engineering 1875. II. P. 218.

***) Zeitschrift des Vereines deutscher Ingenieure. 1887. P. 449.

Comparison of the two.

3) The English horse-power is rather greater than the metrical one, as

1 *IP* Engl. $= 550$ *ft℔s.* per second $= 76.041$ smk

1 *IP* metr. $= 75$ smk $= 542$ *ft℔s.* per second.

To couvert English horse-power into metrical and vice versâ the following coefficients must therefore be used,

1 *IP* Engl. $= 0.9863$ *IP* metr.,

1 *IP* metr. $= 1.0139$ *IP* Engl.

This is however scarcely ever done in practice, for as the difference is not quite 1.4 %, the two horse-powers are regarded as equal.

Kinds of horse-power.

4) The power developed in the cylinder of a steam engine is expressed in horse-power. It is usual to distinguish, according to the method of the calculation,

I) the theoretical horse-power *IP.*,
II) „ indicated „ *IIP.*,
III) „ effective „ *EIP.*,
IV) „ nominal „ *NP.*

Definition.

5) **I. The theoretical horse-power** serves as a basis for calculating the dimensions of a proposed engine, and is in fact the power anticipated from the engine when completed.

Theoretical horse-power of single-expansion engines.

6) *The theoretical horse-power of one cylinder of a single-expansion engine* is found by multiplying the actual mean pressure p_0 per □cm by the *effective piston-area* O in □cm and further multiplying the load,

$$O p_0 \text{ kg},$$

thus produced, by the path of the piston in metres per second, — the *piston-speed* c, and dividing the product

$$O p_0 c \quad \text{mk}$$

by 75

$$IP = \frac{O p_0 c}{75} \quad \ldots\ldots\ldots\ldots\ldots \quad (123)$$

This expression must again be multiplied by x, the number of cylinders, if the total theoretical horse-power of the engine is required, —

$$IP = \frac{O p_0 c x}{75} \quad \ldots\ldots\ldots\ldots\ldots \quad (123_a)$$

Effective piston-area.

7) *The effective piston-area* O, when D cm is the diameter of the cylinder, is

for a trunk-engine with d_T diameter of trunk,

$$O = \frac{\pi}{4}(D^2 - d_T^2) \text{ □cm}; \quad (124)$$

for a piston where the piston-rod and tail-rod have the same diameter d_x,

$$O = \frac{\pi}{4}(D^2 - d_x^2) \ \square\text{cm};$$

when the piston-rod and tail-rod have unequal diameters $d_{,}$ and $d_{,,}$,

$$O = \frac{\pi}{4}\left(D^2 - \frac{d_{,}^2 + d_{,,}^2}{2}\right) \ \square\text{cm}; \qquad (124)$$

when the diameter of the piston-rod is d_x and there is no tail-rod,

$$O = \frac{\pi}{4}\left(D^2 - \frac{d_k^2}{2}\right) \ \square\text{cm}.$$

In general therefore

$$O = \frac{\pi}{4}(D^2 - d^2).$$

8) *The piston-speed* c, for a stroke of H m and N revolutions per minute, **Piston-speed.**

$$c = \frac{2HN}{60} \text{ m per second} \quad \ldots\ldots (125)$$

It is usually

in the oldest marine engines	1 to 2 m
„ later „ „	2 to 3 m
„ recent „ „	3 to 4 m
„ torpedo-boats and fast-running marine engines	4 to 6 m.

9) *The theoretical total horse-power* of a single-expansion engine with x cylinders, becomes, after simplifying Eqq. 124 and 125 **Theoretical total horse-power of single-expansion engines.**

$$IP = 0.000349\ (D^2 - d^2)\ HNp_0 x \quad \ldots\ldots (126)$$

10) *The theoretical horse-power of a multiple-expansion engine* is likewise expressed by Eq. 126, where **Theoretical horse-power of multiple-expansion engines.**

D is the diameter of the LP cylinder, or if there are several of these, D_r must be put instead of D by Eq. 117, (p. 244);

d is the diameter of the LP piston-rod or tail-rod, as per Eq. 124, or d_r the reduced rod diameter, taken instead of d when there are several LP cylinders by Eq. 117,

p_0 the reduced actual mean pressure according to § 29, 18, p. 246;

$x = 1.$

11) **II. The indicated horse-power** is only the work done by the steam in the cylinders, without reference to the losses arising from resistances in the machinery. In well-designed engines the indicated is equal to the theoretical horse-power, but in any case it affords a measure of comparison between the anticipated (theoretical) and actual performance of the engine. It **Definition.**

is now customary to express the power of marine engines almost exclusively in *IHP*.

Indicated mean pressure p_i.

12) The indicated horse-power is found by substituting in Eq. 126, instead of the calculated mean pressure p_o, the *indicated*, i. e. the *measured* mean forward pressure p_i. This pressure is ascertained by means of the indicator, — to be more closely described in a later division. This instrument describes a diagram of energy, the area of which is equal to the work done by the steam per □cm of piston-area during one stroke. The mean breadth of this diagram gives the *mean indicated* pressure p_i.

Determination of the mean indicated pressure.

13) The mean breadth of the indicator diagram is found as follows;

a) *Approximately* by construction, when the cut-off is not too early, by producing the mean ordinate of the admission pressure downwards, increasing its length by $\frac{p}{4}$ and drawing from the point A, thus obtained as centre, an arc of a circle with $^5/_4\, p$ as radius. If the portion BC of the length of the diagram cut-off by this circle is divided so that $BD : BC :: h : H$, and a perpendicular to the atmospheric line is drawn through D, then the portion DE of this perpendicular which is within the circle gives the mean pressure (Fig. 4, Pl. 8). By the process just described, it is in this case $p_i = 1.38$ kg.;

b) *by the usual practical method* which consists in dividing the diagram into 10 equal parts by means of the parallel ruler belonging to the indicator, and measuring the middle height of each of these 10 parts by the scale corresponding to the spring used. The arithmetical mean of these 10 heights also gives p_i; in Fig. 6, Pl. 8 p_i is found to be 1.36 kg by this method on the same diagram;

c) *More accurately* by dividing the length of the card into an even number n of equal parts and calculating the mean breadth of the card by Simpson's Rule

$$p_i = \frac{1}{3\,n}\left[p_o + p_n + 4\,(p_1 + p_3 + \ldots . p_{n-1}) + 2\,(p_2 + p_4 \ldots . p_{n-2})\right]$$

or, less closely thus

$$p_i = \frac{0{,}5\,(p_o + p_n) + p_1 + p_2 + \ldots\ldots p_{n-1}}{n}$$

On Fig. 5, Pl. 8 the mean pressure of the same diagram as computed by Simpson's Rule is $p_i = 1.375$ kg;

d) *Most accurately and rapidly* by means of AMSLER-LAFFON'S planimeter, as is usual in the different Navies. In this way we get p_i for the same diagram $= 1.385$ kg. The

instrument cannot be further discussed here, a full description of it is given in the work named below.*)

14) If the *IHP* of a single-expansion engine with several cylinders, or of a multiple-expansion engine is required, diagrams must be taken simultaneously from all the cylinders, the mean pressure p_i measured, and the **Calculation of the indicated horse-power.**

$$IHP = 0.000349\,(D^2 - d^2)\,H\,N\,p_i \quad \ldots\ldots\ldots \quad (127)$$

for each cylinder determined. The sum of these is the indicated horse-power of the engine.

15) **III. The Effective Horse-power** is, speaking generally, the useful performance of a steam-engine, and is obtained by deducting from the indicated horse-power all the power lost through resistances in the engine itself. The effective horse-power is therefore to be defined as the power transmitted by the crankshaft, or delivered through the first coupling of the shafting of a marine engine. However in the methods of determining the engine-power for a proposed steamer introduced by the elder FROUDE and described in the following division, it has of late become usual to designate as the effective horse-power of a marine-engine no longer the power transmitted by the crankshaft as stated above, but the power actually absorbed in overcoming the ship's resistance alone. So that the effective horse-power of a marine-engine is found by subtracting from its indicated horse-power not only the power lost in resistances in the engine itself, but also in the shafting and propeller as well as in the retarding action of the latter upon the ship's progress. **Definition.**

16) **The effective horse-power of small engines,** such as auxiliary and steam launch engines is determined by a dynamometer, the simplest form of which is the *Prony brake* (Fig. 1, Pl. 9), giving the useful performance of the engine as frictional work. Upon the crankshaft of the engine a sheave *a* is keyed, carrying upon its circumference two wooden brake-blocks *b b*, connected by bolts with the lever *c d*. These bolts are tightened up until the engine runs steadily at the required number of revolutions. At the end *d* of the lever a scale-pan depends, in which are placed sufficient weights to balance the friction of the brake and keep the lever horizontal. The lever must be weighted at *c* so as to lie quite level over the sheave *a* when there are no weights in the scale-pan. Two guards *e* and *f* are fitted, to prevent the lever departing too far from a horizontal position. A stream of water must be run upon the brake-blocks to keep them from firing under the heavy friction. **Effective horse-power of auxiliary engines.**

*) NEHLS: Ueber den Amsler'schen Polarplanimeter etc. Leipzig. 1874.

If N = the number of revolutions of the engine per minute,
l = the length of the lever in m, and
G = the weight in the pan in kg,
then $G\,l$ is the moment which balances the moment of friction $R\,r$ acting at the radius r of the sheave. As the circumferential velocity of the sheave is $\frac{2\,r\,\pi\,N}{60}$, the work expended in friction is

$$\frac{2\,\pi\,N}{60}\,R\,r \text{ mk.}$$

and the corresponding horse-power $\frac{2\,\pi\,N}{60 \times 75}\,R\,r$ HP.

But as $R\,r = G\,l$, we get

$$EHP = \frac{2\,\pi\,N}{60 \times 75}\,G\,l = 0.001396\,N\,G\,l \quad \ldots\ldots \quad (128)$$

Strap-brakes.

17) It is true that the Prony brake is easy both to construct and to use, but strap-brakes are preferable as they occupy less space and are not so subject to oscillation. If therefore it is requisite to make numerous observations of the greatest possible accuracy, the use of strap-brakes rather than block-brakes must be recommended. GRASHOF *) in his "Maschinenlehre" describes a number of modern improved strap-brakes, the best being BRAUER'S, the application of which is there fully discussed, but must here be passed over for want of space.

Froude's determination of the effective horse-power of screw engines.

18) **The effective horse-power of larger marine engines** cannot be ascertained by the above method on account of the great size of the brakes which would be necessary. As before stated, the system introduced by the elder FROUDE **) is mostly adopted. He classifies the resistances which together oppose a screw engine as follows:

a) The nett resistance of the ship, the power absorbed in overcoming which is the effective horse-power *EHP.*;
b) The augmentation of the above resistance, in consequence of the negative pressure exerted by the action of the screw upon the after body of the ship;
c) The water-friction of the screw;
d) The friction of the engine running empty;
e) The increase of this friction due to the resistance of the ship and propeller;
f) The resistance of the pumps.

The sum of these 6 forces expressed in kg, multiplied by the

*) F. GRASHOF. Theoretische Maschinenlehre. Hamburg 1883. Vol. II. P. 832.
**) W. FROUDE. On the ratio of indicated to effective horse-power. Transactions of the institution of naval architects. London 1876. P. 165.

ship's speed in m per second and divided by 75, expresses the power consumed in propelling the ship; FROUDE calls this quantity *the ship's horse-power SIP*. Of course its separate elements can be expressed as fractions of the *EIP*, as this is, in a certain sense, the origin of them all.

19) In order to get at the indicated horse-power from the ship's horse-power, the work done in producing the slip of the screw must be taken account of. And that this also may be expressed as a function of the *EIP*., it is necessary to regard it (the work of slip) as depending upon the speed of the ship, and not, as it really does, upon the speed of the screw. Looked at in this way the different elementary forces from *b* to *f* appear as augmentations of the ship's resistance which would all exist if no slip were present, whereas the slip assumes the form of an increase of force affecting all these elements alike. In other words, the slip necessitates a certain extra number of revolutions per minute of the engine with its total load, both useful and augmentative. **Slip of the screw.**

20) FROUDE determined the negative pressure on the after body caused by the action of the screw from his experiments with models towed in his tank, and found it to be 0.4 of the nett resistance of the ship. The water friction of the screw as deduced from his towing experiments with the "Greyhound" was 0.1 of that resistance. He further assumed, in accordance with earlier views, that the friction-work of the engine, empty and loaded respectively, is in each case $^1/_7$ of the full duty of the engine, and adopted TREDGOLD'S estimate of the resistance of the air and feed pumps, viz. $^1/_{10}$ to $^1/_{20}$ of the total engine resistance, or on an average 0.075 of it. He thus obtained **General magnitude of the elementary forces.**

a) The power absorbed in overcoming the nett resistance of the ship . . . . 1.000 *EIP*
b) the power absorbed in overcoming the negative pressure upon the after body due to the action of the screw . . . . 0.400 „
c) the power absorbed in overcoming the water-friction of the screw . . . . 0.100 „
d) the power absorbed in overcoming the friction of the engine empty . . . . 0.143 *SIP*
e) the power absorbed in overcoming the extra friction of the engine when loaded . . . . 0.143 „
f) the power absorbed in overcoming the friction of the pumps . . . . 0.075 „

$$SIP = 1.5\ EIP + 0.361\ SIP.$$

$$0.639\ SIP = 1.5\ EIP$$

$$SIP = 2.347\ EIP.$$

To this must be added 0.1 *SIP* for the slip and we then get

$$IHP = 1.1\ SHP = 2.582\ EHP$$

$$EHP = 0.387\ IHP \quad \ldots\ldots\ldots\ldots (129)$$

Improvement on the above formula.

21) In order to make this formula, which relates to the *highest* speed V, applicable also to any other speed v, FROUDE separated the power absorbed in overcoming the friction of the engine when running empty (a function of the speed) from the rest of the expression, and substituted for the coefficient 2.582 which he considered somewhat hypothetical, the value 2.7, as being more closely in accordance with practical experience. Of this 2.7 *EHP*, which equals the *IHP* of the engine at the highest speed V, $^1/_7 = 0.385$ *EHP* is, as assumed in 20), absorbed in driving the empty engine, so that the remaining 2.315 *EHP* must be taken up by the other resistances. The *IHP* of the empty engine at any other speed v is proportional to this other speed and is therefore expressed by $\frac{v}{V}\,0.385\ EHP$.

We then get

$$IHP = 2.315\ EHP + 0.385\,\frac{v}{V}\,EHP \quad \ldots\ldots\ldots (130)$$

the *EHP* being that which belongs to the speed V. At the end of his paper FROUDE expressly says that the above formula is not to be taken as a perfect one, but only as a well grounded step in the right direction.

Magnitude of the different elements of the indicated horse-power.

22) It thus appears, by these assumptions of FROUDE'S that the losses of *IHP* in a screw engine are

α) through friction of the engine, — empty . .	0.130 *IHP*
β) through extra friction of the engine loaded .	0.130 *IHP*
γ) through the work of the pumps	0.069 *IHP*
δ) through water-friction of the screw	0.039 *IHP*
ε) through negative pressure on the vessel's after body	0.155 *IHP*
ζ) through slip	0.090 *IHP*
together	0.613 *IHP*,

so that only 0.387 *IHP* remains, as actually employed in propelling the ship; see Eq. 129.

General magnitude of the efficiency η of marine engines.

23) RAUCHFUSS*) found, after analysing the trial-trip results of 56 ships of the German navy (mostly comparatively old) that, on full-power runs

$$EHP = \text{from } 0.35 \text{ to } 0.45\ IHP.$$

On runs taken at less than full-power, these coefficients are

*) E. RAUCHFUSS. Widerstand und Maschinenleistung der Dampfschiffe. Kiel 1886. P. 34.

somewhat increased, because, according to Eq. 130, the friction of the empty engine diminishes with the speed and therefore the useful power is greater. — The efficiency, $\eta = \frac{EIP}{IIP}$, called by FROUDE *the propulsive coefficient,* for the German war-ships investigated by RAUCHFUSS, is given in the following table; it rises, as is well known, with the size of the ship, and therefore also with that of the engines.

Table of the Efficiency of Marine-engines.

Displacement of the ship in tons	Value of η mean	approximate.
1	2	3
8000 and above	0.460	0.470
4000 to 8000	0.455	0.450
2000 to 4000	0.435	0.430
1000 to 2000	0.404	0.410
500 to 1000	0.402	0.390
200 to 500	0.381	0.370
200 and below	0.368	0.350

24) It may here be remarked that recent experiments with towed models by DENNY and the younger FROUDE, as well as progressive trials carried out by the former, have shewn for large modern steamers of ordinary form

Efficiency η from more recent research.

$$EIP = \text{from } 0.45 \text{ to } 0.50\ IIP \quad \ldots\ldots\ldots \quad (130)$$

The latter value is only exceeded in the newest and extraordinarily fine-lined steamers, where as high a result as

$$EIP = \text{from } 0.53 \text{ to } 0.55\ IIP$$

has been reached. In one particular case — that of a steamer built by INGLIS, having a separate engine to drive the circulating pump, DENNY calculated that the $EIP = 0.6\ IIP$; it is also stated that in one of YARROW'S fast torpedo boats, with the air, circulating, and feed pumps separately driven and the corresponding duty excluded from the IIP of the main engines, the EIP was even $= 0.75\ IIP$. These are however such very exceptional cases hitherto that they hardly affect the value of DENNY'S rule for good average ships, viz.

$$EIP = 0.50\ IIP.$$

25) **Later correction of FROUDE's figures.** In recent years many experiments have been made, the results of which materially modify the numerical values in 22) for the different elements of resistance to a marine engine, as based upon FROUDE'S assumptions. These new values explain of themselves how it is that FROUDE'S

Modification of Froude's numerical values.

propulsive coefficient of 0.387 has become enlarged into DENNY'S of 0.50, and they shall therefore be briefly stated.

Work of the empty engine.

26) *a. The work developed by marine engines when running empty* has been repeatedly determined by Mr. MUDD, Manager of the Central Marine Engineering Co., West-Hartlepool. He fitted up a foundation plate in the erecting-shop on which the engines were not only put together, but tried under steam. The results of two of his experiments of this sort have been published, one with the triple engines of the S. S. "Cleveland" *), the other with the compound engines of the S. S. "Stepney" **).

Experiment with the engines of the "Cleveland".

27) Fig. 2, Pl. 9 shews the indicator cards of the "Cleveland's" engine under these circumstances and running at 63 revolutions. The distribution was as follows

In the H P. cylinder	. . . . . .	16.10 *IHP*
„ „ M P. „	. . . .	5.13 „
„ „ L P. „	. . . .	23.10 „
	together	44.33 *IHP*.

As the engines indicate about 900 horse-power when at work (a sister-engine in the S. S. "Enfield" ***) shewed a mean of 866 *IHP* on 4 measured-mile runs), the power of the empty engine is only about 5 % of the indicated power at sea. In the experiment in the shop, the circulating pump had to draw the water a horizontal distance of 137 m and lift it 5.94 m, corresponding to 5.5 *HP*, to which were added 5.5 *HP* for friction in the contracted water mains, making 11 *HP* together; so that only 34 *IHP*, or about 4 % of the full power is left as due to driving the empty engine. This would have to be increased by about 2 to $2^1/_2$ % of the full *IHP*, for driving the shafting empty, so that the total empty power would be 6 to $6^1/_2$ % of the indicated horse-power at sea.

Experiment with the engines of the "Stepney".

28) The two-cylinder compound engine of the S. S. "Stepney", running empty at 80 revolutions with the wheel at full-gear and the throttle-valve nearly shut, produced the cards shewn in Fig. 3, Pl. 9, giving 54.8 *IHP*. After deducting 6 to 7 horse-power absorbed by the circulating pump, we find the power required to drive the empty engine is about 6 to 7 % of the full *IHP* of 650. MUDD inferred from the comparison of these two engines that the three-cranked engine absorbed in running empty about 2 to $2^1/_2$ % of its full *IHP* less than the two cranked engine. If we take into consideration that the resistance of the empty engine when perfectly new and only just

*) Engineering 1886. I. P. 206.

**) Engineering 1886. II. P. 290.

***) Engineering 1886. I. P. 10.

erected must be greater than that of a well kept engine which has been some time at work, we must conclude that the assumption of 13 % of the full *IHP* for this resistance must be at least twice too high, and that in first class engines of several thousand horse-power, it will be as low as 5 %.

29) The engines tried by ISHERWOOD, to which we shall return in the next division, shewed a mean resistance when running empty of 8 % of their full *IHP*. They are mostly small engines of less than 100 *IHP* and it is an old experience that the work absorbed in moving these is proportionately greater than in very large engines. It may be mentioned that ISHERWOOD found the empty resistance of small launch engines to be in some cases as low as 5 and even 3 % of their full *IHP*. *It thus appears from the foregoing that we shall be on the safe side in assuming the empty resistance of well kept marine engines of medium size to be 7 % of their full IHP.* **Isherwood's experiments.**

30) *β. The extra work of friction* appears, from some very careful experiments of THURSTON'S*) not to exist at all in engines which are well kept and lubricated. In three engines tried by him, of 20, 100, and 20 *IHP* the resistances of the engines themselves when empty and loaded were in all three cases equal and amounted respectively to 12 %, 8.88 %, and 9.62 % of the full *IHP*. An *augmentation* of friction due to the load on the engine did *not* occur, so that the power absorbed in overcoming its friction remained constant from the empty condition to that of full power. It is true that in a fourth engine the work of friction varied between 13.5 and 17.5 *HP*, but this was entirely caused by irregularities in the lubrication. THURSTON'S results agree very well with ISHERWOOD'S and as both refer to small engines they only serve to corroborate the concluding sentence of 29). ISHERWOOD, in his investigations, which were made before THURSTON'S, always assumes the extra frictional work of the loaded engine to be 7.5 % of the full *IHP*, or rather more than half FROUDE'S allowance only. **Thurston's experiments.**

31) BRAUER,**) in his experiments with five portables and a semi-portable compound condensing engine, found the friction of the fully loaded engines to be greater than when they were running empty, except with portable No. 3 where the contrary was the case. In portables Nos. 1 and 2 it first increased and afterwards decreased and only rose steadily in portables Nos. 4 and 5, and the compound engine. On comparing the results **Experiments of Brauer and others.**

*) Journal of the Franklin Institute. Philadelphia. 1888. November Number.

**) Der Civilingenieur. Leipzig 1884. Numbers 4 and 5.

of these experiments with THURSTON'S, and bearing in mind what extreme care and attention are necessary in taking indicator cards under such circumstances in order to detect the very small differences in the powers given off during the application of the brake, we cannot altogether reject THURSTON'S assertion that *the work due to friction in an engine is the same whether it runs empty or loaded; that is, the extra friction due to the load is zero.*

Table of the frictional work of the various parts.

Parts	Engine I 20 *IHP* Slide balanced	Engine I 20 *IHP* Slide unbalanced	Engine III 20 *IHP*	Engine II 100 *IHP*	Engine IV 70 *IHP*
1	2	3	4	5	6
Main bearings	47.1	35.4	35.0	41.6	46.0
Piston and piston-rod	32.9	25.0	21.0	49.1	21.0
Crank-pin	6.8	5.1	13.0		
Cross-head	5.4	4.1			
Slide	2.5	26.4	22.0	9.3	21.0
Excentrics	5.3	4.0			
Link and excentrics	—	—	9.0	—	
Air-pump	—	—	—	—	12.0
Together	100.0	100.0	100.0	100.0	100.0

Frictional work of the various parts.

32) The percentage portions contributed by the various parts of the engines to the total frictional work, as determined by THURSTON and given in the above table, are very noteworthy. It appears that the main bearings account for the greatest share, viz. from 35 to 47 % of the total frictional work, or 4 % of the full *IHP* of the engines as given in 30). Second in importance is the friction of the piston and rod, the lowest figure for which is 21 % of the total friction or 2 % of the *IHP*. This proportion may probably be reduced by the use of a good metallic packing for the piston-rod gland and close attention to the piston. The third item is the friction of the unbalanced slide-valve, which under some circumstances, exceeds that of the piston; the other frictional resistances are of less importance.

Work of the pumps.

33) *γ. The work of the pumps* was likewise taken much too high by FROUDE. As an instance of the truth of this assertion, at any rate when the pumps are driven by separate engines, we may quote the trial-trip results of several modern twin-

screw cruisers, the first of which belongs to the German and all the others to the U. S. Navy.*)

Table of the work of the pumps of modern cruisers.

Ship's name	"Princess Wilhelm"	"Philadelphia"	"San Francisco"	"Newark"	"Yorktown"	"Concord"
1	2	3	4	5	6	7
Date of trial	Jan. 1890	June 1890	August 1890	Decr. 1890	Feb. 1889	Jan. 1891
IHP of the main engines	9241	8533	9587	8582	3510	3314
" " " air pump engines	258	65	108	62	126 (air pump and circ.)	27 (air pump and circ.)
" " " circ. " "	48	55	29	30		
" " " feed " "	34	34	43	53	16	18
Gross *IHP*	9571	8687	9761	8727	3642	3359
Percentage work of the air pump engines	2.591	0.747	1.103	0.705	3.453 (air pump and circ.)	0.813 (air pump and circ.)
" " " " circ. " "	0.501	0.633	0.301	0.346		
" " " " feed " "	0.355	0.397	0.439	0.607	0.443	0.525
" " " " pumps (total)	3.447	1.777	1.843	1.658	3.896	1.338

So that the whole resistance of the pumps would in these engines account for 2.33 % only on an average, and not 7.5 %. There is no reason to assume that the work of the pumps, when separately driven as is now preferred, would be materially greater in other engines, so that 4 % appears to be quite sufficient to allow for it until further experience is gained.

34) δ. *The work absorbed in overcoming the water friction of the screw* appears, unlike the items hitherto discussed, to have been taken decidedly too low by FROUDE. Screws larger than the comparatively small one of the "Greyhound" and working, as they do in fast ships, at a much higher number of revolutions, must absorb far more power in water friction than 4 % of the *IHP* of the engines, as FROUDE assumes. ISHERWOOD found the water friction of the small screws he tried, to average from 8 to 9 % of the *nett IHP* of the engines (gross *IHP* less the frictional work of the empty engine), so that 10 % of the full *IHP* must be regarded as a moderate estimate for large fast steamers. **Water friction of the screw.**

35) ε. *The negative pressure exerted upon the after body by the action of the screw* was measured by FROUDE first and afterwards by TIDEMANN**) in the course of their experiments with towed models. The results of his experiments published by the latter confirm in general the approximate correctness of FROUDE'S **Negative pressure on the after body.**

*) Journal of the American Society of Naval Engineers. Washington 1889—1891.

**) B. J. TIDEMANN. Memorial van de Marine. Amsterdam 1876—80. Part. II. P. 75—90.

figures, viz 15 % of the full *IHP* as the loss of power due to the unfavourable action of the propeller upon the after body.

The work of slip.

36) ζ. *The work of slip* was probably correctly estimated by FROUDE at about 9 % of the *IHP*. ISHERWOOD, who, in his experiments combined the work of the negative pressure on the after body and that of slip together as one quantity, found its magnitude to be on an average 22 % of the *nett* power (see 34) of the engine, which about corresponds to from 20 to 21 % of the *IHP*. If the work of slip is taken at 9 %, there remains, as due to the screw's action upon the after body from 11 to 12 % of the *IHP*, against the 15 % assumed by FROUDE. The difference between these values need cause no surprise, as FROUDE experimented with the "Greyhound", whose lines were comparatively full and ISHERWOOD with small fast steamers of fine after body. Besides, the estimation of this particular percentage is based rather upon the personal experience of the experimenters than upon direct observation.

Compilation of the results of recent research.

37) On tabulating the various elements of resistance as determined from the above-mentioned experiments, we arrive at the following result, viz.

Empty work of engine	0.070 *IHP*
Extra work due to load	0.000 „
Work of the pumps	0.040 „
Water-friction of the screw	0.100 „
Loss of work through action of screw on run	0.150 „
Work of slip	0.090 „
Together	0.450 *IHP*.

so that we have remaining

$$EHP = 0.550\ IHP,$$

for overcoming the nett resistance of the ship, a value which has been already reached in large fast steamers.

Possible improvement of the propulsive coefficient η.

38) The question suggests itself whether a further increase of the efficiency of marine engines is likely to be attained to. In view of the inventive energy displayed by designers, this question must be decidedly answered in the affirmative. We think it has been shewn in the preceding that even now, superior engines of large size, in which there is no extra frictional resistance when loaded, require but little more than 8 % of their full *IHP* for engine friction and work of pumps together, although these two items are put at 11 % in the last table. Efforts have been made to diminish the friction of propellers by polishing the surfaces of the blades and sharpening their edges; the effect of the screw's action upon the run is obviated to a certain extent by making the after body very

long and fine and placing the screw as far aft as can possibly be managed; and the problem of the reduction of the loss from slip is being attacked by paying very great attention to the diameter, pitch and form of propeller. It is therefore to be hoped that in future the average propulsive coefficient of good engines of large size will reach and even exceed 0.60.

39) **IV. Nominal horse-power** is an expression for the power of an engine which has become obsolete, but in WATT'S time it was identical with the indicated horse-power. For low-pressure engines there were several, widely different methods of calculating it, but it at length became a mere ratio of the piston-area, which was here and there applied to compound engines, but is no longer used for the triples of the present. Nominal horse-power.

40) Nearly all WATT'S low-pressure engines had an indicated mean pressure of 7 lbs. per □″ Engl. = 0.492 kg per □cm. He proportioned the piston speed of his engines to the stroke H (in feet Engl.) so that $128\sqrt[3]{H}$ was the piston speed in feet per minute, or $0.966\sqrt[3]{H}$ metres per second, when the stroke is given in metres. Watt's formula.

The expression for nominal horse-power thus took the following form, neglecting the sectional area of the piston-rod

$$NP = \frac{\frac{\pi}{4} D^2 \times 0.966 \sqrt[3]{H} \times 0.492 \times x}{75}$$

$$NP = 0.00498\, D^2 \sqrt[3]{H}\, x \quad \ldots\ldots\ldots\ldots (131)$$

41) The British Admiralty afterwards altered this formula by substituting for WATT'S obsolete piston-speed the actual piston-speed of the engine, and thus introduced the so-called British Admiralty formula.

$$\text{Admiralty } NP = \frac{\frac{\pi}{4} D^2 \times 2\,HN \times 0.492 \times x}{75 \times 60}$$

$$\text{Admiralty } NP = 0.000171\, D^2 H N x \quad \ldots\ldots\ldots (132)$$

In British units this formula becomes

$$\text{Admiralty } NP = \frac{\frac{\pi}{4} D^2 \times 2\,HN \times 7 \times x}{33000} \quad \ldots\ldots (132^a)$$

$$\text{Admiralty } NP = \frac{D^2 \times 2\,HN \times x}{6000} \quad \ldots\ldots\ldots (132^b)$$

where D is the diameter of piston in inches,
$2\,HN$ is the piston speed in feet per minute,
and x is the number of cylinders.

42) If we substitute in Eq. 132^b the piston-speed $2\,HN = 200$ feet Engl. per min., we get Simplified Admiralty formula.

$$NP = \frac{D^2 \times 200 \times x}{6000} = \frac{D^2 \times x}{30} \quad \ldots\ldots\ldots (133)$$

This formula has been most commonly used in the British Mercantile Marine. It means that if the piston area is regarded as a square of the side D in inches Engl., then 30 such square inches (or "circular inches" as they were called) are necessary to produce one NP. For D in cm the above formula becomes

$$NP = \frac{D^2\, x}{193.54} \quad \ldots\ldots\ldots (133^a)$$

American formula.

43) In the United States Navy, the following modification of the British Admiralty formula was adopted,

$$NP = \frac{(D-1)^2 \times 2\,H\,N \times x}{5640}, \quad \ldots\ldots\ldots (134)$$

which gives a greater NP for large engines and a lesser one for small engines than the British Admiralty formula.

Penn's formula.

44) In the German Navy the NP of screw engines was determined by means of PENN'S formula, in which the piston-speed is taken at 360′ per min. = 1.828 m per sec., — thus

$$NP = \frac{\frac{\pi}{4} D^2 \times 0.492 \times 1.828 \times x}{75}$$

$$NP = 0.0093\, D^2\, x \quad \ldots\ldots\ldots (135)$$

Formula for compound engines.

45) For compound engines the British formula (133) was afterwards altered to

$$NP = \frac{D_H^2 + D_N^2}{33} \quad \ldots\ldots\ldots (136)$$

in which

D_H is the diameter of the H. P. cylinder in inches Engl.

D_N „ „ „ „ „ L. P. „ „ „ „

or the "reduced" diameter of the L. P. cylinders where there are more than one.

With the diameters given in cm, the formula becomes

$$NP = \frac{D_H^2 + D_N^2}{212.9} \quad \ldots\ldots\ldots (136^a)$$

Proportion of the nominal to the IHP.

46) The low indicated mean pressure and piston-speed at which WATT'S engines worked compared with those of modern practice, account for the nominal horse-power not exceeding the fifth or sixth part of the indicated horse-power of a recent engine. In latter years, before the British Admiralty gave up nominal horse-power, it was there regarded as in fact representing one sixth of the IHP for compound engines; and in the French Navy an order was issued in 1867 that the nominal horse-power was to be taken as one fourth of the indicated. The reason why the term "nominal horse-power" was so long retained is

partly because the weights of marine engines were often determined by the nominal horse-power, but chiefly because it was the custom to regulate the price of engines on that basis.

47) The great commercial importance of this latter consideration, and the difficulty of fixing upon a uniform measure of engine power for use in the shipping registers induced the interested circles in England to occupy themselves repeatedly with the problem of devising a new Rule for engine power in accordance with the progress of mechanical engineering. This question was first advanced by the Board of Trade at the Institution of Naval Architects in 1872*) and the Institution limited itself to recommending the indicated horse-power as the most suitable criterion of the performance of an engine. The choice of the indicated power is certainly the most obvious one, but renders the taking of indicator cards indispensable. This is however not always done, as it is quite a matter of indifference to many owners what power their engines develop, provided they work very economically, i. e. that they propel a ton of cargo one knot or 100 knots at a certain speed with the smallest consumption of coal. This circumstance moved the North-east Coast Institution of Engineers and Shipbuilders in 1888**) to frame a formula enabling the "normal *IHP*" of ordinary triples to be calculated from their dimensions. This formula is not given here, as it was not widely adopted. It has been justly remarked that a rule of this sort, even if it gives the average indicated power of the present triple expansion engine with fair exactness, will soon be left behind in the perpetual progress of engineering and the same state of things will then again be reached as in the days of nominal horse-power. — In the Navy Lists of the various Powers the engines are almost invariably entered with their indicated horse-power. **Normal indicated horse-power.**

§ 31.

Consumption of steam.

1) We may distinguish **Definitions.**

I) the theoretical steam consumption D_t,

II) the indicated steam consumption D_i, and

III) the actual steam consumption D.

The symbols D_t, D_i and D are to refer to the hourly steam

*) Transactions of the institution of naval architects. London 1872. P. 340.

**) Transactions of the north-east coast institution of engineers and shipbuilders. Newcastle upon Tyne. 1887—88. P. 335.

consumption of an engine. With the suffix IP they will signify the hourly steam consumption per IP, — thus, $D_{t_{IP}}$, $D_{i_{IP}}$, D_{IP}.

Theoretical steam consumption.

2) **I. The theoretical steam consumption** is found as follows. If 1 kg of perfectly dry steam $(x = 1)$ expands adiabatically from the admission pressure p down to the condenser pressure p_1, then its dryness fraction x_1 at the end of the expansion is determined by the first equation on p. 41

$$x_1 = \frac{a + b\,x - a_1}{b_1}$$

where a_1 and b_1 have values corresponding to p_1. The external work performed by the kg of steam, in T. U. is by Eq. 46

$$A\,L = q - q_1 + x\,\varrho - x_1\,\varrho_1 \quad \text{T. U.}$$

By § 30, 1

$$1\ IP = 75 \text{ mk per second}$$

$$1\ IP = 75 \times 3600 = 270000 \text{ mk per hour, or}$$

$$1\ IP = \frac{75 \times 3600}{424} = 637 \text{ T. U. per hour.}$$

So that the theoretical weight of steam per IP per hour is

$$D_{t_{IP}} = \frac{637}{A\,L} \text{ kg} \quad \ldots\ldots\ldots\ldots\ldots \quad (137)$$

and this multiplied by the number of horse power of the engine is of course its theoretical hourly steam consumption.

Example.

3) For a single-expansion engine where the admission pressure $p = 3$ atmos., the condenser pressure $p_1 = 0.2$ atmos., and $x = 1$, we find by the table on p. 41 that

$$x_1 = \frac{0.3993 + (1.2645 \times 1) - 0.1984}{1.6975} = 0.86$$

and thereupon by the steam-table, p. 28

$$A\,L = 133.85 - 60 + (1 \times 470.30) - (0.86 \times 528) = 90 \text{ T. U.}$$

Consequently the hourly theoretical steam consumption per IP of this engine is

$$D_{t_{IP}} = \frac{637}{90} = 7.077 \text{ kg.}$$

Comparison.

4) Calculating in the same way the theoretical steam consumption for a compound engine of the most usual admission pressure $p = 5$ atmos., also for a modern triple with $p = 12$ atmos. and finally for a quadruple with $p = 15$ atmos., the condenser pressure being assumed the same $p_1 = 0.2$ atmos. in each case, we obtain the following values

Single-expansion	engine,	$p = 3$ atmos.,	$p_1 = 0.2$ atmos.,	$D_{t_{IP}} = 7.077$ kg
Compound	„	$p = 6$ „	$p_1 = 0.2$ „	$D_{t_{IP}} = 5.687$ „
Triple-expansion	„	$p = 13$ „	$p_1 = 0.2$ „	$D_{t_{IP}} = 4.697$ „
Quadruple „	„	$p = 16$ „	$p_1 = 0.2$ „	$D_{t_{IP}} = 4.445$ „

5) The theoretical steam consumption per *HP* per hour as determined above is an ideal one, never reached in reality, as it could only be possible if **Smallest steam consumption.**

1) the steam were perfectly dry, which is only very rarely the case,
2) there were no losses from cooling and leaks which can never be entirely avoided,
3) the combined diagram exactly coincided with the theoretical one circumscribed about it, which is impossible for the reasons detailed in § 29, 17.

6) **II. The indicated, or useful steam consumption** is determined from the average indicator card (§ 18, 13) of all the cylinders of a single-expansion engine or from the H. P. cylinder of a compound or multiple-expansion engine. The first step is to find by inspection (see Fig. 7, Pl. 8) the point of cut-off 2 (§ 18, 8) and also the point 0 at which the compression ends, or in other words, the admission begins. The ordinates p_e and p_c of these two points represent the absolute pressures at the moments of cut-off and of admission. From these pressures the weight γ_e and γ_c respectively of a cbm of the steam can be found in the steam table on p. 28. The volume of the steam admitted to the cylinder per stroke multiplied by γ_e, minus the volume remaining in the cylinder at the end of the compression multiplied by γ_c, gives the indicated weight of steam used per stroke. **Indicated steam consumption.**

7) To determine the volume admitted to the cylinder per stroke the ordinates of the two extreme ends of the card are drawn; then the portion of the axis of abscissæ which lies between these two ordinates corresponds to the stroke, *H* (metres) and also to the volume swept (cubic metres). A length, to the same scale of volumes, is now to be measured off beyond the foot of the admission ordinate, equal to the volume of the clearance. If this is regarded as the m'th part of the volume swept, its length is of course $= m' H$ (see § 18, 19). The difference between the abscissa of the point 2 and that of the extreme admission end of the card, corresponds to $h = \varepsilon H$, the portion of the stroke completed at the moment of cut-off, so that **Volume of steam at cut-off.**

$$m' H + \varepsilon H = H(\varepsilon + m') \text{ metres}$$

represents, *on the scale of volumes*, that portion of the cylinder which is filled with steam of pressure p_e at the moment of cut-off. Further, if

$D =$ the diameter of the cylinder in cm,
$d =$ „ „ „ „ piston rod &c. (see Eq. 124)

then the total volume at the moment of cut-off is expressed by

$$\frac{\frac{\pi}{4}(D^2 - d^2)}{10000} \times H(\varepsilon + m') \text{ cbm}$$

and its weight by

$$\frac{\frac{\pi}{4}(D^2 - d^2)}{10000} \times H(\varepsilon + m')\,\gamma_e \text{ kg.}$$

Volume of steam at the moment before admission.

8) At the moment before admission the clearance space $m'H$ is filled with steam of pressure p_c. Where the slide has much lead and opens the port very early, the piston has generally only reached the point 0 when the admission begins. The abscissa $h_c = \varepsilon_c H$, of this point plus the distance $m'H$ corresponding to the clearance

$$m'H + \varepsilon_c H = H(\varepsilon_c + m') \text{ m}$$

represents that portion of the cylinder which, immediately before the admission begins, is still filled with steam of pressure p_c, the volume of which is therefore

$$\frac{\frac{\pi}{4}(D^2 - d^2)}{10000} \times H(\varepsilon_c + m') \text{ cbm}$$

and its weight

$$\frac{\frac{\pi}{4}(D^2 - d^2)}{10000} H(\varepsilon_c + m')\,\gamma_c \text{ kg.}$$

Weight of steam per hour.

9) The weight of fresh steam per stroke consequently is

$$\frac{\frac{\pi}{4}(D^2 - d^2)}{10000} H\Big[(\varepsilon + m')\,\gamma_e - (\varepsilon_c + m')\,\gamma_c\Big] \text{ kg.}$$

and for N revolutions per minute or $120\,N$ strokes per hour we get

$$D_i = \frac{\frac{\pi}{4}(D^2 - d^2)}{10000} H\Big[(\varepsilon + m')\,\gamma_e - (\varepsilon_c + m')\,\gamma_c\Big]\, 120\,N \text{ kg}$$

$$D_i = 0.009425\,(D^2 - d^2)\,HN\Big[(\varepsilon + m')\,\gamma_e - (\varepsilon_c + m')\,\gamma_c\Big] \text{ kg.} \quad (138)$$

The above formula gives the indicated hourly steam consumption for *each* cylinder of a single-expansion engine and for a compound or a multiple expansion engine where D cm is the diameter of the H. P. cylinder.

Simplification of the above formula for single-expansion engines.

10) For marine engines the abscissa h_c of the point 0 is always so small that the indicator card very seldom shews it, and thus ε_c may usually be neglected and Eq. 138 somewhat simplified. If we wish to simplify it still further for the sake of convenience, we can (by § 18, 51) regard SC Fig. 10, Pl. 2 as that portion

of the clearance which must be filled with fresh steam at every stroke, under the assumption that this steam serves to raise the pressure at the end of the compression p_c to the admission pressure p. The expression within the brace then assumes the following form for a single-expansion engine, bearing in mind what is said in § 27, 3, —

$$(\varepsilon + m')\,\gamma_e - (\varepsilon_c + m')\,\gamma_c = (\varepsilon + m)\,\gamma_e = \varepsilon'\,\gamma_e;$$

and we get

$$D_i = 0.009425\,(D^2 - d^2)\,H\,N\,\varepsilon'\,\gamma_e \text{ kg} \quad \ldots\ldots \quad (138^a)$$

Simplified formula for multiple-expansion engines.

11) To make the preceding applicable to multiple-expansion engines, we must substitute ε_g' as the actual total ratio of cut-off, and correspondingly instead of D the diameter of the H. P. cylinder, D_N that of the L. P. cylinder or D_r the reduced diameter (Eq. 117), — and γ_e must be taken to represent the specific weight of the steam at the beginning of the expansion in the L. P. cylinder. The indicated steam consumption per hour is then calculated from

$$D_i = 0.009425\,(D_N^2 - d^2)\,H\,N\,\varepsilon_g'\,\gamma_e \text{ kg} \quad \ldots\ldots \quad (138^b)$$

It must be expressly borne in mind that the last two formulæ can only give approximate values. Where *accuracy* is required formula 138 must always be used.

12) For purposes of comparison it is usual to determine not only the total indicated steam consumption per hour of the engine, but also the corresponding quantity per horse-power. For a single-expansion engine this is obtained by dividing Eq. 138[b] by Eq. 127, which, taken in the approximate form for the sake of simplicity, gives,

Hourly indicated steam consumption *IP*.

$$D_{i_{IP}} = \frac{0.009425\,(D^2 - d^2)\,H\,N\,\varepsilon'\,\gamma_e}{0.000349\,(D^2 - d^2)\,H\,N\,p_i} = \frac{27\,\varepsilon'\,\gamma_e}{p_i} \text{ kg} \quad \ldots \quad (139)$$

As the indicated horse-power of a multiple-expansion engine is obtained by substituting in Eq. 127 the diameter D_N or D_r of the L. P. cylinder and the indicated pressure p_{i_r} reduced to the L. P. piston, as given in § 30, 10, we may express the hourly steam consumption per *IP* by

$$D_{i_{IP}} = \frac{0.009425\,(D_N^2 - d^2)\,H\,N\,\varepsilon_g'\,\gamma_e}{0.000349\,(D_N^2 - d^2)\,H\,N\,p_{i_r}} = \frac{27\,\varepsilon_g'\,\gamma_e}{p_{i_r}} \text{ kg} \quad \ldots \quad (139^a)$$

13) The hourly indicated steam consumption per horse-power may be estimated or predicted from the corresponding theoretical consumption, if we take into consideration that the latter does not in fact produce the full theoretical work as shewn on the theoretical diagram, but only that due to the actual indicator diagram inscribed in it. Therefore the weight of steam consumed per horse-power must be increased in the same proportion as the indicated work is less than the theoretical, for instead of

Predicting the indicated steam consumption.

producing a horse-power, the kg of steam as calculated in 4) only do the βth part of one. This leads to

$$D_{i_{IP}} = \frac{D_{t_{IP}}}{\beta} \text{ kg} \quad \ldots \ldots \ldots \ldots \quad (140)$$

As the values given on pp. 247 to 249 for the coefficient β shew, this coefficient is taken very high at 0.66 for single-expansion engines and 0.70 for multiple-expansion engines. We should then obtain by Eq. 140 the following estimated or predicted steam consumption per IP per hour, viz. for

Single-expansion engines,	$p = 3$ atmos.;	$p_1 = 0.2$ atmos.;	$D_{i_{IP}} = 10.72$ kg
Compound „	$p = 6$ „	$p_1 = 0.2$ „	$D_{i_{IP}} = 8.12$ „
Triple-expansion „	$p = 13$ „	$p_1 = 0.2$ „	$D_{i_{IP}} = 6.71$ „
Quadruple „	$p = 16$ „	$p_1 = 0.2$ „	$D_{i_{IP}} = 6.35$ „

These values are however always exceeded in reality because they are calculated, like the theoretical ones in 5), without any allowance for moisture of the steam, cooling, and leakages.

Basis of this method.

14) **WARRINGTON'S method***) of determining the indicated steam consumption has the advantage of enabling us to find it for a single-expansion engine with no information as to its dimensions and piston-speed, but simply from the indicator cards. This method starts from the assumption that the engine uses, instead of steam, water or any other incompressible fluid of 1 atmos. pressure and cuts off at the end of the stroke. Thereupon the constant 27 in Eq. 139 and 139ᵃ) becomes 26.12734, because the pressure of one atmosphere upon a sq. metre is 10334 kg, γ_e becomes $= 1000$, as 1 cbm of water weighs 1000 kg, further p_i or $p_{i_r} = 1$, and if the clearance is neglected, ε' and $\varepsilon'_g = 1$ likewise. The water used per horse-power per hour is then

$$26.12734 \times 1000 = 26127.34 \text{ kg};$$

and if the water had been under p'_e atmos. pressure instead of 1 atmos.

$$\frac{26127.34}{p'_e} \text{ kg}.$$

Extension of this principle to steam.

15) If the engine be now supposed to use, instead of water, steam of the same pressure, the volume of fluid used will be the same for the same power, but the weight of the steam will be less than that of the water in the ratio of their respective specific gravities. If v'_e cbm of steam at p'_e atmos. pressure are produced from 1 kg of water, the weight of this volume of steam is to the weight of an equal volume of water as $1 : 1000\, v'_e$ because v'_e cbm of water weigh 1000 v'_e kg. Under these assumptions the weight of steam per horse-power per hour will be

*) CH. T. PORTER. A treatise on the Richards steam-engine indicator etc. London 1875.

$$W = \frac{26127.34}{1000\, p'_e\, v'_e} \text{ kg.}$$

16) If we next assume that the indicated mean pressure p_i is equal to the terminal pressure p'_e, or in other words that the specific volume v'_e corresponds to the pressure p_i, the steam consumption for any absolute terminal pressure can be calculated. If the indicated mean pressure p_i differs from the terminal pressure p'_e (as it of course invariably does), the indicated steam consumption will be proportionately increased, assuming the specific weight of steam to be proportional to the pressure, which is nearly correct, where the range of pressures is not too great. The indicated consumption per *IP* per hour may thus in general be expressed by **Relation to the mean indicated pressure.**

$$\frac{W p'_e}{p_i} \text{ or } \frac{W p'_e}{p_{i_r}} \text{ kg.}$$

The table on the following page gives the values of v'_e, W, and $W p'_e$ for various terminal-pressures p'_e.

17) The indicated steam consumption as calculated by the preceding must be corrected for clearance and compression if it is to be even approximately accurate. This correction is only possible when the volume of the clearance is known, and the corresponding distance $m'H$ set off upon the mean indicator diagram (Fig. 7, Pl. 8) as described in 7), making the total length of the diagram $H + m'H = H(1 + m')$. The expansion line must now be produced by the eye beyond the point of release to the end of the diagram, as the dotted curve shews. The ordinate of this end of the curve is the terminal pressure p'_e. If now the mean indicator card is drawn in again but in the reverse direction as dotted, WARRINGTON says that the ratio of the length $H(1 + m')$ to the sum H' of the length representing the clearance and the abscissa measured in the direction of the piston's motion of that point on the reversed card at which the compression pressure equals the terminal pressure p'_e, is also the ratio of the steam consumption as above calculated to the steam consumption as corrected for clearance and compression. So that the corrected steam consumption is **Effect of compression and clearance.**

$$\frac{W H'}{H(1 + m')}$$

and therefore the indicated consumption per *IP* per hour

$$D_{i_{IP}} = \frac{W H' p'_e}{H(1 + m')\, p_i} \text{ kg.} \quad \ldots\ldots\ldots\ldots (141)$$

18) WARRINGTON'S process cannot be used for multiple-expansion engines if the indicator cards are the only information available. For such engines, besides the cards, at least the dimensions **Applicability of this process.**

Table of indicated steam consumption per *IP* per hour according to Warrington.

Absolute terminal pressure in Atm. p'_e	Specific volume of the steam in cbm v'_e	$p'_e v'_e$	$W = \frac{26127.34}{1000 p'_e v'_e}$ kg	$W p'_e$	Absolute terminal pressure in Atm. p'_e	Specific volume of the steam in cbm v'_e	$p'_e v'_e$	$W = \frac{26127.34}{1000 p'_e v'_e}$ kg	$W p'_e$
1	2	3	4	5	1	2	3	4	5
0.1	14.504	1.450	18.010	1.801	3.90	0.458	1.786	14.63	57.057
0.2	7.525	1.505	17.418	3.483	4.00	0.447	1.788	14.61	58.440
0.3	5.128	1.540	16.960	5.088	4.10	0.437	1.792	14.58	59.778
0.4	3.908	1.560	16.750	6.700	4.20	0.427	1.793	14.56	61.152
0.5	3.165	1.580	16.530	8.265	4.30	0.418	1.797	14.53	62.479
0.6	2.665	1.600	16.339	9.803	4.40	0.409	1.799	14.52	63.888
0.7	2.304	1.610	16.230	11.361	4.50	0.400	1.800	14.51	75.295
0.8	2.031	1.620	16.120	12.896	4.60	0.392	1.803	14.49	66.654
0.9	1.818	1.630	16.020	14.418	4.70	0.384	1.805	14.45	67.915
1.0	1.646	1.646	15.870	15.870	4.80	0.377	1.810	14.43	69.264
1.1	1.505	1.655	15.780	17.385	4.90	0.370	1.813	14.41	70.609
1.2	1.386	1.663	15.710	18.852	5.00	0.363	1.815	14.39	71.950
1.3	1.285	1.670	15.640	20.332	5.10	0.356	1.816	14.38	73.338
1.4	1.199	1.680	15.540	21.756	5.20	0.350	1.820	14.36	74.672
1.5	1.123	1.684	15.510	23.265	5.30	0.343	1.821	14.35	76.055
1.6	1.057	1.691	15.450	24.720	5.40	0.337	1.823	14.33	77.382
1.7	0.999	1.699	15.370	26.129	5.50	0.332	1.825	14.31	78.705
1.8	0.946	1.703	15.340	27.612	5.60	0.326	1.826	14.30	80.080
1.9	0.899	1.708	15.290	29.051	5.70	0.321	1.829	14.26	81.282
2.0	0.857	1.714	15.243	30.486	5.80	0.316	1.833	14.25	82.650
2.1	0.819	1.718	15.208	31.937	5.90	0.311	1.835	14.24	84.016
2.2	0.784	1.725	15.146	33.321	6.00	0.306	1.836	14.23	85.380
2.3	0.751	1.727	15.128	34.794	6.25	0.294	1.838	14.21	88.812
2.4	0.722	1.733	15.076	36.182	6.50	0.284	1.845	14.16	92.040
2.5	0.695	1.741	15.002	37.505	6.75	0.273	1.848	14.13	95.377
2.6	0.670	1.742	14.990	38.974	7.00	0.265	1.855	14.10	98.700
2.7	0.646	1.744	14.970	40.190	7.25	0.256	1.856	14.07	100.997
2.8	0.625	1.750	14.929	41.801	7.50	0.248	1.860	14.04	105.300
2.9	0.604	1.752	14.921	43.271	7.75	0.241	1.867	13.99	108.422
3.0	0.586	1.758	14.861	44.583	8.00	0.234	1.872	13.96	111.680
3.1	0.568	1.761	14.838	45.998	8.25	0.227	1.873	13.95	114.077
3.2	0.551	1.763	14.818	47.417	8.50	0.221	1.878	13.91	118.235
3.3	0.535	1.765	14.790	48.807	8.75	0.215	1.881	13.89	121.537
3.4	0.521	1.771	14.749	50.146	9.00	0.209	1.883	13.86	124.740
3.5	0.507	1.774	14.723	51.330	9.25	0.204	1.887	13.84	128.020
3.6	0.493	1.775	14.720	52.992	9.50	0.199	1.891	13.81	131.195
3.7	0.481	1.780	14.680	54.316	9.75	0.194	1.893	13.80	135.550
3.8	0.469	1.782	14.660	55.708	10.00	0.190	1.900	13.75	137.500

of the cylinders must be given, in order to determine the indicated pressure p_{i_r}, which is to be substituted instead of p_i in the above formula, and p'_e must be taken to represent the terminal pressure in the L. P. cylinder. But even for single-expansion engines this method only gives approximate values unless the clearance is known and the correction for it (see 17) can be applied. Simple as WARRINGTON'S process is, its utility in marine work is thus very limited.

19) **III. The actual steam consumption** can only be accurately determined by a water-meter inserted in the feed-pipe, registering the quantity of feed-water supplied to the boilers during a certain time. Assuming the boiler to be tight and the water-level to be the same at the beginning and end of the test, the weight of steam used in a certain time must equal the weight of water fed into the boilers in the same time, provided the water is free from any matter which is separated and precipitated during evaporation. If such matter is present in any appreciable quantity its weight must be found by analysis and deducted from that of the water, to get the actual weight of steam. **Exact determination of the actual steam consumption.**

20) *Water-meters* as applied to boilers were reported upon at the nineteenth meeting of engineers connected with the International Union of Boiler Supervising Associations from the 11th. to the 14th. of July 1890 by Mr. STRUPLER,*) Superintendent Engineer — in reply to the question put forward by the Union: what experience have the various associations had as to the water-meters used by them? Reports had come to hand from eleven associations on SCHMID'S meter, from eight on KENNEDY'S, from four on SIEMENS'S, from two on DREYER, ROSENKRANZ, and DROOP'S, from one on SCHÄFFER AND BUDENBERG'S and from one on FISCHER AND STIEHL'S. **Kinds of water-meters.**

21) According to these reports, the pistons of SCHMID'S and KENNEDY'S meters are subject to rapid wear causing their accuracy to diminish in the course of time. SIEMENS'S apparatus is more durable, but less exact. Its error is occasionally as high as 20 % and it is easily clogged with dirt. Krupp obtained very satisfactory results with FISCHER AND STIEHL'S meter on the screw principle. But this apparatus cannot be used under pressure and is besides heavy, expensive, and takes up considerable space. The piston meters of SCHMID and KENNEDY are the most used and when applied to stationary boilers have in general given satisfactory measurements. SCHMID'S requires **Reports on the water-meters.**

*) Zeitschrift des Vereines deutscher Ingenieure. 1890. P. 960.

repairs at rare intervals only, in fact hardly any, if carefully looked after. In KENNEDY'S the packing is sensitive. The error of SCHMID'S is from 0.5 to 1.5 % and of KENNEDY'S from 2 to 3 %. Unfortunately these more perfect meters are not adapted for continuous working.

Trial trip experience with Schmid's meter.

22) SCHMID'S meter has until quite lately been repeatedly used on the trial-trips of German war vessels, but the results were only middling, as the errors caused by the ship's oscillations and by the air contained in the feed-water were too great. The utility of this apparatus would probably not be much increased even if the air and grease were separated from the feed-water in a heater intercepting it on its way to the meter, as STRUPLER states that the latter can stand no higher temperature than 70° C. If hotter water has to be temporarily used, extra lubrication is said to assist the meter's action, but it is doubtful if this remedy could be extended over a trial trip lasting several hours. Water-meters will not be further described here on account of their unsuitability for use afloat. Their various constructions are treated of in the periodical referred to below.*)

Approximate determination of the actual steam consumption.

23) *The actual steam consumption may be approximately calculated* from the indicated or useful steam consumption and the unavoidable losses. These losses are due to *cooling* of the steam in the pipes and on the cylinder walls as well as to *leaks* at the joints and the various parts of the engine which confine the steam such as glands, slides, and particularly pistons. For both these losses, but with reference to the cylinders alone HRABÁK**) and KÁŠ have developed empirical formulæ, which are briefly mentioned below, as GRASHOF***) states that they appear to be nearer the truth than any other formulæ introduced hitherto.

Items of the loss from cooling.

24) **The loss from cooling** D_a arises in the first place from the external radiation of an unjacketed cylinder (see § 17, 23) and from the communication of heat from the jacket steam to the working steam in a jacketed one (see § 17, 22). The loss accounted for in this way may be judged of approximately by help of the experiments made with a range of steam-pipes (referred to in § 25, 2). On the other hand, the estimation of the greater part of D_a, viz. the loss from initial condensation due to the influence of the cylinder walls upon the entering steam, is still based upon

*) Zeitschrift des Vereines deutscher Ingenieure. 1874. PP. 145 & 427.

**) J. HRABÁK. Hilfsbuch für Dampfmaschinen-Techniker. Berlin 1881. Section II. P. 101.

***) F. GRASHOF. Theoretische Maschinenlehre. Hamburg 1890. Vol. III. P. 695.

very uncertain assumptions, in spite of the recent efforts of many investigators in the direction of its more exact determination as described in § 17, 30 to 35.

25) HRABÁK and KAŠ, in developing their formula, start from the hypothesis that the loss from cooling is in the first place proportional to the range of temperature between the entering and exhausting steam, that is approximately to the difference $p - p'$, if p' is the exhaust pressure. This loss is further influenced by the extent of metallic surface in contact with the steam during admission, which may be taken as about equal to half the internal surface of the cylinder plus the areas of the cylinder cover and piston = Derivation of the formula.

$$0.5\,\pi\,D\,H + 2 \times 0.25\,\pi\,D^2 = 0.5\,\pi\,D\,(D + H).$$

Finally, the weight of steam per stroke, determined by the actual ratio of cut-off ε' (see p. 238) must be taken into account, so that the absolute loss due to cooling in a certain time may be expressed by

$$\mathrm{D}_a = x\,D\,(D + H)\,\varepsilon'\,(p - p')\ \mathrm{kg} \quad \ldots \ldots \ldots \quad (142)$$

Investigations have shewn that if D_a represents the loss from cooling in the cylinder per hour, the following substitutions must be made in this equation which relates only to single expansion engines, viz.

D = the diameter of the cylinder in metres,

x = 370 to 400.

For compound engines the equation takes the form

$$\mathrm{D}_a = x\,D_N(D_N + H)\,\varepsilon_g'\,(p - p_1)\ \mathrm{kg}, \quad \ldots \ldots \quad (142^a)$$

where D_N = the diameter of the L. P. cylinder in metres

p_1 = the terminal pressure of the L. P. cylinder in atmos.

x = 300 to 400;

the other terms have the same signification as before (for ε_g' see p. 238).

26) One defect of the above formula is that it gives the same loss from cooling for a jacketed as for an unjacketed cylinder. Value of the formula.
In consequence of the assumption that the reduction of the loss from cooling is balanced by the consumption of jacket steam, the ratio of this loss to the power developed by the engine may be pretty accurately determined, as HRABÁK calculates it to be greater in jacketed engines, although the loss D_a, due purely to cooling, which is here in question, must be less in jacketed than in unjacketed cylinders and this circumstance ought to be brought into account by the coefficient x. Further, the mere taking of x between certain limits does not sufficiently regard the influence of the piston-speed c upon the loss. The shorter the time during which the entering steam

remains in contact with the cooler cylinder wall, the more rapidly does the heat penetrate the latter, so that the more strokes occur in a certain time, i. e. the greater c becomes, the more D_a must increase. According to § 16, 8 and § 17, 33 however, D_a does not rise in the same ratio as c, hence, although the proportion of the loss D_a to the useful steam consumption D_i, or in other words the loss per *IHP* per hour, diminishes as c increases, still this is not true to the extent of its being inversely proportional to c, which the above formula assumes to be case.

Derivation of the formula.

27) **The loss from leakage** D_u, especially from that at the piston, was formerly regarded by VOELCKERS as constituting the principal portion of the total loss of steam, whereas according to recent experience, it is comparatively slight in good engines. According to VOELCKERS*) the mean loss from leakage per second was

$$D_u = 0.00132\, D \sqrt{p_i} \text{ kg}$$

where D in cm is proportional to the periphery of the piston, i. e. the length of the space passing steam, and $\sqrt{p_i}$ to the working pressure — or the velocity with which the steam escapes. The amount of play between the piston and cylinder is supposed to be the same for engines of all sizes. This gives the loss much too high for small engines and too low for large ones. HRABÁK accordingly improved upon VOELCKERS'S formula, which may in general be written $D_u = a\, D \sqrt{p_i}$, by the introduction of a further factor $b\, D^2 p_i$ which increases in a higher ratio than D, because when once the piston is in a leaky condition the amount of play is, under similar circumstances, greater in a large engine than a small one. As by Eq. 123 we may put

$$\frac{IHP}{c} = \frac{O\,p_i}{75} = \frac{\frac{\pi}{4} D^2 \times p_i}{75},$$

$$\frac{IHP}{c} \text{ is proportional to } D^2 p_i$$

and the hourly loss from leakage may be expressed as

$$D_u = a \sqrt{\frac{IHP}{c}} + b\,\frac{IHP}{c} \text{ kg.} \qquad (143)$$

The constants a and b are

for single-expansion engines $a = 17.6$ and $b = 1.0$

" compound " $a = 12.3$ and $b = 0.7$.

Inapplicability of both formulæ.

28) Even assuming that the formulæ 142 and 143 are sufficiently accurate for single expansion and compound engines up to

*) J. VOELCKERS. Der Indicator. Berlin 1878. Edit. II. P. 66.

9 atmos. absolute admission pressure and 5 m piston speed, they would still not be adapted to the triple expansion marine engine of to-day with its much higher working pressure. They are besides open to the objections raised in 26) with regard to the calculation of the loss from cooling in the cylinder, and they altogether leave out of account the loss from cooling in the pipes. For all these reasons it is preferable to make use of the actual results of experiments carried out with the object of determining the gross loss of steam, than to attempt to calculate it in detail for modern marine engines from HRABÁK'S formulæ. The former method has become more and more usual in practice as the impossibility of taking all the various sources of error, mostly of a purely local and accidental character, into account in an exact theoretical formula is increasingly evident.

Estimation of the actual steam consumption.

29) *The actual steam consumption* can be estimated by help of the results of the following experiments which were made for this purpose, viz.

a) the trials both of older and more modern German war-ships, to which further reference will be made later on.
b) EMERY'S trials of American steamers (see the tables, pp. 84 to 88),
c) WIDMANN'S trials of French war-ships (see the tables pp. 120 and 121.
d) RISBEC'S*) experiments with the French cruisers "Chasseur" and "Voltigeur".
e) KENNEDY'S**) trials of the British merchant steamers "Fusi Yama", "Colchester", "Meteor" and "Tartar".

Although certain objectious may be raised to the method of conducting most of these trials, they afford data for the determination of values which cannot differ widely from the average results obtained in practice.

Value of the coefficient ξ.

30) It is to be inferred from the above that, for moderately good marine engines *when developing their utmost power*, the useful steam consumption D_i only amounts to the following fraction ξ of the gross, or actual steam consumption D:

$\xi = 0.70$	to	0.75	for	single-expansion	engines	without jackets
$\xi = 0.75$	„	0.80	„	„	„	with „
$\xi = 0.75$	„	0.80	„	compound	„	without „
$\xi = 0.80$	„	0.85	„	„	„	with „
$\xi = 0.80$	„	0.85	„	triple-expansion	„	without „
$\xi = 0.85$	„	0.90	„	„	„	with „

*) Memorial du génie maritime. 1882. No. V.

**) Engineering. 1889 I. P. 527 and 1890 I. P P. 577, 605 and 632.

The value of ξ for triples will also be nearly correct for good quadruples of at least 16 atmos. absolute boiler pressure, of which no experimental data are as yet available.

General minimum value of $D_{i_{IP}}$

31) Thus the actual hourly steam consumption is obtained by dividing Eq. 138 by ξ, — thus

$$D = \frac{D_i}{\xi} \text{ kg and } D_{IP} = \frac{D_{i_{IP}}}{\xi} \text{ kg} \quad \ldots\ldots\ldots \quad (144)$$

The actual steam consumption per horse-power per hour *at the utmost power of the engines* and corresponding to the values given for the indicated consumption (see p. 270) is now in round numbers, for

Single-expansion engines	$p = 3$ atmos.,	$p_1 = 0.2$ atmos.,	$D_{IP} = 12.0$ kg
Compound „	$p = 6$ „	$p_1 = 0.2$ „	$D_{IP} = 9.0$ „
Triple-expansion „	$p = 13$ „	$p_1 = 0.2$ „	$D_{IP} = 7.5$ „
Quadruple „ „	$p = 16$ „	$p_1 = 0.2$ „	$D_{IP} = 7.0$ „

In compound and multiple-expansion engines, designed and constructed with particular care, working at the most favourable ratio of expansion and with dry steam, so that their coefficient of fulness β exceeds 0.70, the actual steam consumption per IHP per hour is below the above respective quantities. Reports of any better results than the above are in general to be received with mistrust, the more so the nearer they approach the figures of p. 270. If they are lower than these even, they must be in any case regarded as exaggerated, unless the indicator diagrams and all other particulars of the trials are published in support of them. It may be remarked that SCHRÖTER*) found the actual steam used per *IHP* per hour in a horizontal stationary triple-expansion engine by the Augsburger Maschinenfabrik with 10.5 atmos. working pressure, cutting off at 0.25 stroke in the H. P. cylinder, to be 5.46 kg at the best, and cutting off at 0.3 stroke 5.79 kg at the worst — the condensation in the pipes being left out of account. If this is added, the above quantities are increased by 2.9 and 4.3 % to 5.62 and 6.04 kg respectively. These results may be regarded as about the best attainable with the very highest class of triple-expansion engines of equal working pressure.

Average working values of $D_{i_{IP}}$

32) The average actual steam consumption per *IHP* per hour of ordinary marine engines, working continuously and not in a state of special preparation for trials, may be taken, *for favourable cut-off ratios*, as follows

6.0 to 7.0 kg in the most recent quadruple-expansion engines at 14 to 15 atmos. boiler pressure;

*) Zeitschrift des Vereines deutscher Ingenieure. 1890. P. 10.

6.5 to 8.0 kg in the most recent triples at 10 to 12 atmos. boiler pressure;
8.0 to 9.0 kg in the older triples at 8 to 10 atmos. boiler pressure;
8.5 to 9.5 kg in recent compound engines at 6 to 7 atmos. boiler pressure;
9.5 to 11.0 kg in older compound engines at 4 to 5 atmos. boiler pressure;
11.0 to 13.0 kg in compound engines at 3 atmos. boiler pressure with surface condensers, jackets, and superheaters;
12.0 to 14.0 kg in low-pressure engines at 2 atmos. boiler pressure with surface condensers, jackets, and superheaters;
14.0 to 16.0 kg in low-pressure engines at 2 atmos. boiler pressure with jet condensers and without jackets or superheaters;
16.0 to 18.0 kg in low-pressure engines at less than 2 atmos. boiler pressure with jet condensers and without jackets or superheaters.

Assuming the engines to be in good order in other respects, the working results approach the more closely to the minimum figures given above, the drier the steam is and the more efficient the jacketing.

33) If D_{IP} kg of steam produce 1 *IHP* throughout 1 hour = 3600 seconds, the same weight of steam must be capable of exerting 3600 *IP* for 1 second. Therefore 1 kg can perform $\frac{3600}{D_{IP}}$ horsepower per second. Among German engineers this value has become known as the "*ratio of excellence*" (Güteverhältniss) of an engine. "Ratio of excellence".

It will be about

600 *IHP* for the best quadruples of the future,
500 „ for the best triples,
400 „ for superior compound engines,
300 „ for medium pressure compound engines,
200 „ for the older low-pressure engines.

These figures exhibit in a striking manner the important progress made in marine engineering in the last thirty years, for the greater number of steamers at work about 1860 had engines of the last-named description.

§ 32.

Quantity of injection and circulating water.

1) **I. Quantity of injection water** W_i. The exhaust steam on leaving the cylinders of a single-expansion engine, or the L. P. cylinder Heat of the exhaust steam.

of a multiple-expansion engine, has, as a rule, a pressure of 0.5 to 0.9 atmos. or about 0.7 atmos. on an average, so that its heat (of the steam) is by the table on p. 29 about 595 T. U.

Heat given up by the steam.

2) This steam is converted into water, the temperature of which is usually about 40^0 in the condenser; so that there must be withdrawn from it per kg

$$595 - t_c = 595 - 40 = 555 \text{ T. U.}$$

Heat taken up by the injection water.

3) If the withdrawal of the heat from the steam is accomplished in a jet-condenser by contact with sea-water, at a temperature which may be taken on an average in our latitudes as 15^0, then every kg of sea-water can take up

$$t_c - t_s = 40 - 15 = 25 \text{ T. U.}$$

before reaching the condenser temperature.

Weight of injection water W_i.

4) Thus the condensation of 1 kg of steam in a jet-condenser requires

$$W_i = \frac{595 - t_c}{t_c - t_s} = \frac{595 - 40}{40 - 15} = \frac{555}{25} = 22.2 \text{ kg} \quad \ldots \ldots (145)$$

of injection water. In order to be safe, it is usual to calculate 25 kg of injection water per kg of exhaust steam.

Heat taken up by the circulating water.

5) **II. Weight of circulating water** W_x. In the case of a surface condenser, the water entering at $t_s = 15^0$ is heated to an average discharge temperature of 25^0, so that it can take up per kg

$$t_a - t_s = 25 - 15 = 10 \text{ T. U.,}$$

Weight of circulating water W_x.

6) therefore, to condense 1 kg of steam in a surface condenser, we require

$$W_x = \frac{595 - t_c}{t_a - t_s} = \frac{595 - 40}{25 - 15} = \frac{555}{10} = 55.5 \text{ kg} \quad \ldots \ldots (146)$$

of circulating water.

Variability of these values.

7) The proper quantity of injection water for any engine at any particular power of which it is capable must finally be decided by actual trial with the vacuum gauge: the above formulæ only give general average values and leave both the influence of latitude upon the sea temperature and also the different designs of condenser out of account.

Temperature of the sea.

8) In tropical waters, where the sea temperature is considerably above 15^0, the necessary weight of injection water rises to 30 kg and that of the circulating water to 70 kg. The mean annual sea temperature is

in our latitudes	11^0 C.
in the Mediterranean . .	18^0 C.
in 20^0 of latitude (N or S)	24^0 C.
on the Equator	28^0 C.

Influence of the arrangement of condenser.

9) The design of a surface-condenser of course determines the number of times the water traverses it and influences the

period of time occupied by the process; the longer this lasts the more heat the water takes up. Experience shews that in temperate latitudes the condensation of 1 kg of steam in a surface condenser requires

in traversing the condenser once with a period of transit of 2 to 3 seconds; $W_x = 50$ to 60 kg;

in traversing the condenser three times with a period of transit of 5 to 6 seconds, $W_x = 40$ kg;

in traversing the condenser eight times with a period of transit of 12 to 13 seconds, $W_x = 26$ kg.

The last named quantity was determined by WIDMANN*) on the trial trips of the French iron-clad "Friedland" launched in 1873. The rise of temperature of the circulating water was from 12^0 to 40^0, while a good vacuum was steadily maintained. Surface condensers with defective circulation may take a weight of circulating water from 80 to 100 times as great as the weight of steam to be condensed and nevertheless only produce a vacuum of 0.8 to 0.88 kg. per sq. cm., as shewn in the trials of the French ironclads "Victorieuse" and "Colbert". In the original condensers of these ships, large dead spaces were formed in which the water remained stationary instead of circulating and thus rendered a considerable portion of the cooling surface ineffective. The remainder of the tube surface was insufficient to produce a good vacuum in spite of an extra weight of circulating water.

10) **III. Leakage water W_l.** Marine engines are almost universally arranged so that in case of a leak, the injection water for jet condensers or the circulating water for surface condensers can be taken from the bilge, in the former case by the air-pump, in the latter by the circulating pump. If the steam used per hour by the engine at full power is D kg (Eq. 144, p. 278) then **Leakage Water.**

$D W_i$ kg or $0.001\ D W_i$ tons (jet)

and $D W_x$ kg or $0.001\ D W_x$ tons (surface)

would require to be pumped from the bilge in the same time. Experience shows however that most engines can no longer develop their greatest power when taking their condensing water from the bilge, because the quantity capable of being supplied from this source is insufficient to produce the highest vacuum.

11) The injection water flows with a certain head from the sea into the condenser, according to the depth below the water- **Leakage water with jet condensers.**

*) E. WIDMANN. Etudes des principes de la construction des machines marines. Paris 1890. P. 106.

line at which the rose is placed. But if the water is to be taken from the bilge, it has to be lifted several feet by the vacuum in the condenser through a long range of pipes with bends, valves, &c. besides having to pass the obstruction of the strum, so that to be on the safe side, only 75 % of the greatest quantity of injection water can be calculated upon. That is, for jet condensers

$$W_l = 0.00075\, D\, W_i \text{ tons} \quad \ldots\ldots\ldots\ldots \quad (147)$$

Leakage Water with surface condensers.

12) The circulating water of surface condensers also flows under a certain hydrostatic pressure from the sea into the circulating pump, which in most modern engines is a centrifugal pump. In ordinary work the water has to be forced only a few feet up. But the quantity delivered by the pump is very considerably diminished when it has to raise the water several feet from the bilge, particularly when the suction pipes, as usual on board ship, have many bends and obstructions. The quantity of leakage water that can be dealt with is therefore only about 66 % of the greatest weight of circulating water on an average, or

$$W_l = 0.00066\, D\, W_x \text{ tons} \quad \ldots\ldots\ldots\ldots \quad (147^a)$$

with surface condensers.

§ 33.

Quantity of brine to be blown off.

Saltness.

1) The feed-water, as it was used before the introduction of the now customary boiler pressure of over 100 lbs., contained a certain quantity of inorganic components in solution, called in general *salt*. This steadily increasing amount of salt in the boiler had to be removed by either constant or periodical blowing-off.

System of feed.

2) **I. Constant blowing-off.** The quantity of brine to be blown off depends, in the first place, upon where the feed is taken from, which may be either from

a) the sea,

b) a jet condenser, or

c) a surface condenser.

Quantity of brine W_a to be blown off when feeding from the sea.

3) a. **Feeding from the sea.** The feed-water contains on an average 1 to 4 % of salt according to the locality, thus every kilo of feed carries in solution $s = 0.01$ to 0.04 kg of salt into the boiler. Consequently when a kg of sea-water is evaporated its s kg of salt remain in the boiler water, the saltness s_x of which is thus gradually increased. In former practice this was not allowed to exceed 7 %. Later on the saltness at working

pressures not over 100 ℓ/s was kept in the German Navy between $7\frac{1}{2}$ and 9 %, in the French between 9 and $10\frac{1}{2}$ %, in the British, when feeding from the sea or from jet condensers, up to 11 %, and with surface condensers as high as 14 %. Now as 1 kg of feed-water in reaching the same concentration as the rest of the water in the boiler, can only take up $s_x - s$ kg of salt, it follows that the salt left in the boiler from 1 kg of *feed-water evaporated*, will saturate as many kg of *feed-water not evaporated* as are expressed by the quotient of $\frac{s}{s_x - s}$. To keep the percentage of salt in the boiler water from exceeding s_x, we must blow off, per kg of feed, the following weight of water which becomes saturated by the evaporation of every kg of feed, viz.

$$W_a = \frac{s}{s_x - s} \text{ kg.} \qquad (148)$$

So that for D kg of steam used per hour (Eq. 144, p. 278), there must be blown off

$$W_{a_1} = D \frac{s}{s_x - s} \text{ kg} \qquad (149)$$

4) This equation may also be derived as follows. The total feed-water per hour is **Another derivation of this formula.**

$$D + W_{a_1} \text{ kg}$$

containing $(D + W_{a_1})\, s$ kg of salt;
there are blown off per hour $W_{a_1} s_x$ kg of salt.
If the percentage of salt is to remain constant, these two quantities of salt must obviously be equal to each other, consequently

$$W_{a_1} s_x = (D + W_{a_1})\, s$$
$$W_{a_1} (s_x - s) = D s$$
$$W_{a_1} = D \frac{s}{s_x - s} \text{ kg.}$$

5) If, for instance as in the Atlantic, the sea water contains 3.5 % of salt, i. e. $s = 0.035$ and the boiler water is to be kept at 7 %, there must be blown off per kg evaporated **Example.**

$$\frac{s}{s_x - s} = \frac{0.035}{0.07 - 0.035} = 1 \text{ kg}$$

so that the quantity blown off is equal to that evaporated:

$$W_{a_1} = D$$

But with 11 % of saltness in the boiler, we only have to blow off

$$\frac{s}{s_x - s} = \frac{0.035}{0.11 - 0.035} = 0.47 \text{ kg}$$

per kg evaporated.

Saltness in jet condensers s_c.

6) b. **Feeding from jet condensers.** It is to be noted that, the percentage of saltness of the injection water being s, the $W_i\,s$ kg of salt introduced into the condenser during the condensation of 1 kg of steam are distributed after the condensation over $(W_i + 1)$ kg of water. So that the water taken from the jet-condenser now only contains a percentage of saltness

$$s_c = \frac{W_i\,s}{W_i + 1} = s\,\frac{\frac{595 - t_c}{t_c - t_s}}{\frac{595 - t_c + t_c - t_s}{t_c - t_s}} = s\,\frac{595 - t_c}{595 - t_s}\text{ kg} \quad \ldots \quad (150)$$

or taking the numerical values of Eq. 145

$$s_c = s\,\frac{595 - 40}{595 - 15} = s\,\frac{555}{580} = 0.95\,s$$

Quantity of brine to be blown off when feeding from jet-condensers.

7) The quantity of brine to be blown off per kg of water evaporated when feeding from a jet condenser is therefore

$$W_a = \frac{s_c}{s_x - s_c}\text{ kg} \quad \ldots\ldots\ldots\ldots \quad (151)$$

and taking the figures in 5) with 7 % of saltness in the boiler,

$$W_a = \frac{0.035 \times 0.95}{0.07 - 0.035 \times 0.95} = 0.9\text{ kg}$$

or with 11 %

$$W_a = \frac{0.035 \times 0.95}{0.11 - 0.035 \times 0.95} = 0.43\text{ kg.}$$

Salt supplementary feed.

8) c. **Feeding from surface condensers.** The boilers require to receive besides the condensed water, the *supplementary feed* rendered necessary by the various losses of steam. With working pressures not exceeding 100 ℔s., this is generally taken from the circulating water, the saltness of which is s. The supplementary feed is about 2 % of the condensed water when the boilers and engines are in good tight condition, but in practice it is assumed to be 5 % for high-pressure boilers, to be on the safe side. If there are any serious leaks, as was sometimes the case with low-pressure boilers, it may of course be considerably more than this. Thus 100 kg of feed water from a surface condenser contain 5 kg of supplementary feed or 5 s kg of salt. In one kg of this feed water there are therefore

$$s_0 = 0.05\,s\text{ kg of salt} \quad \ldots\ldots\ldots\ldots \quad (152)$$

Fresh supplementary feed.

9) Here we may mention that in modern warships and merchant steamers having double bottoms, several compartments of these are often filled with fresh water to be used for supplementary feed. It is however more usual to fit a special evaporator for distilling the supplementary feed water, to be described later on. With such an arrangement brining of course becomes unnecessary.

Quantity of Brine to be blown off when feeding from surface condensers.

10) When feeding from a surface condenser and without the above mentioned arrangement, there must be blown off per kg of water evaporated

$$W_a = \frac{s_0}{s_x - s_0} \text{ kg} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (153)$$

For instance with 3 % of salt in the sea and 7 % in the boiler

$$W_a = \frac{0.05 \times 0.035}{0.07 - 0.05 \times 0.035} = 0.025 \text{ kg}$$

and with 14 %

$$W_a = \frac{0.05 \times 0.035}{0.14 - 0.05 \times 0.035} = 0.013 \text{ kg.}$$

With old low pressure engines whose boilers are not quite tight the quantity of supplementary feed may rise until there is $^1/_2$ % of salt in the feed, or s_0 becomes $= 0.005$. In such a case W_a increases to 0.077 kg with 7 % of salt in the boilers or 0.039 with 14 %.

Measuring the saltness.

11) Before the above calculations can be made for a boiler at work, the saltness of the water must in each case be first determined by the salinometer.

Practical advantage of periodical blowing.

12. **II. Periodical blowing.** It was already observed with the old low-pressure boilers where continuous brining was necessary that there was a difficulty in preventing the formation of a current of water direct from the check-valve to the blow-off. If this could not be stopped, alternate feeding and blowing had to be resorted to, in order not to immediately eject the fresh feed instead of the brine from the boiler. After the introduction of surface condensers, periodical blowing off was more and more adopted for the above reason, and is now the universal practice on account of the very small amount of brining required with modern high-pressure boilers; although it is theoretically more correct to blow continuously, as will be shewn later in 19).

Saltness of the boiler water after blowing.

13) When blowing periodically the quantity of brine W'_a to be removed at each operation is calculated as follows.

Let $G =$ the weight in kg of boiler water at half glass,
$D' =$ „ „ „ „ „ water evaporated during the period between two blowings,
$W'_a =$ the weight in kg of water blown out each time,
$s_x =$ „ „ „ „ „ salt per kg of boiler water at the time of blowing.

It is assumed that immediately before blowing, the average weight G of the boiler water is increased by 0.5 W'_a kg to $G + 0.5\ W'_a$ kg, and that after blowing there are $G - 0.5\ W'_a$ kg

left in the boiler, so that at that moment the total weight of salt in the boiler is

$$(G - 0.5\ W''_a)\ s_x \text{ kg.}$$

Saltness s_x at the time of blowing.

14) If now there are fed in 0.5 W''_a kg of saltness s, to bring the water again to half glass, the weight of salt rises to

$$(G - 0.5\ W''_a)\ s_x + 0.5\ W''_a\ s \text{ kg.}$$

And if, before the next blowing, D' kg are evaporated and replaced by feed, thus bringing the saltness to the limit s_x, then the weight of salt in the boiler is

$$G\,s_x = (G - 0.5\ W''_a)\ s_x + 0.5\ W'_a\ s + D'\ s.$$

If again, just before blowing, 0.5 W''_a kg of feed water are introduced, to keep the water at the necessary height, the weight of salt is

$$(G - 0.5\ W''_a) s_x + 0.5\ W''_a s + D'\ s + 0.5\ W''_a s = (G - 0.5\ W''_a) s_x + (W''_a + D') s \text{ kg.}$$

From what has been said, we must have

$$(G + 0.5\ W'_a)\ s_x = (G - 0.5\ W''_a)\ s_x + (W''_a + D')\ s \text{ kg.}$$

$$\left[(G + 0.5\ W'_a) - (G - 0.5\ W'_a)\right] s_x = (W''_a + D')\ s$$

whence follows the saltness at the time of blowing

$$s_x = \frac{W'_a + D'}{W'}\ s \quad \ldots \ldots \ldots \ldots \ldots \quad (154)$$

Weight of brine to be blown off W'_a

15) Substituting the value of s_x (Eq. 154) in the expression for $G\,s_x$ in 14), we get

$$G\,s_x = (G - 0.5\ W'_a)\left(\frac{W'_a + D'}{W'_a}\right) s + 0.5\ W'_a\ s + D'\ s$$

$$G\,s_x = G\,s\left(\frac{W'_a + D'}{W'_a}\right) - 0.5\ W'_a\,s - 0.5\ D'\,s + 0.5\ W'_a\,s + D'\,s$$

$$G\,s_x = G\,s\left(\frac{W'_a + D'}{W'_a}\right) + 0.5\ D'\ s$$

$$G\,s_x = G\,s\left(1 + \frac{D'}{W'_a}\right) + 0.5\ D'\ s \text{ kg}$$

whence follows the proportion of the water evaporated to that blown off:

$$\frac{D'}{W'_a} = \frac{G\,s - 0.5\ D'\,s}{G\,s} - 1$$

$$\frac{D'}{W'_a} = \frac{G\,s_x - G\,s}{G\,s} - \frac{D'}{2\,G}$$

$$\frac{D'}{W'_a} = \frac{s_x - s}{s} - \frac{D'}{2\,G} \quad \text{und}$$

$$W'_a = \frac{G\,D'\,s}{G\,(s_x - s) - 0.5\ D'\,s} \text{ kg} \quad \ldots \ldots \ldots \quad (155)$$

In this equation we must write s, s_c, or s_o according to the source from which the feed-water is taken.

Continuous brining economically better than periodical.

16) The boilers will be most economically worked when the smallest possible quantity of water is blown off, and this is the case

when D′ is zero, in other words, when they are blown continuously instead of periodically, as will be shewn by an example in 19).

17) When the boiler is filled with sea-water, containing $G s$ kg of salt, and has been a certain time at work, receiving per minute D_m kg of feed of saltness s, it will reach the permissible saltness s_x in t minutes, and then necessarily **Interval between blowing.**

$$G s + t D_m s = G s_x$$

whence follows

$$t = G \frac{s_x - s}{D_m s} \text{ Minutes, } \quad \ldots\ldots\ldots\ldots (156)$$

the period at the end of which the first blowing must take place. There again s, s_c, or s_0 must be employed according to circumstances.

18) A large low-pressure boiler contains $G = 20000$ kg of water of $s = 0.035$ kg saltness. It is fed with water of the same saltness and is to be blown at a limiting saltness of $s_x = 0.07$ kg. The boiler steams an engine of 500 *IHP*, requiring 14 kg of water per *IHP* per hour, so that the weight of feed-water per minute $D_m = \frac{500 \times 14}{60}$ kg, and therefore **Example for the time to elapse before the first blowing.**

$$t = 20000 \times \frac{0.07 - 0.035}{0.035} \times \frac{60}{500 \times 14}$$

$$t = 171 \text{ Minutes.}$$

19) After this, Eq. 155 enables us to determine the weight of brine that must be periodically blown off. Assuming the blow-off cock to be opened every 10 minutes, we should have the weight of feed for this period **Example for the quantity of brine to be removed when blowing periodically.**

$$D' = \frac{500 \times 14 \times 10}{60} = 1167 \text{ kg}$$

and therefore

$$W'_a = \frac{20000 \times 1167 \times 0.035}{20000\,(0.07 - 0.035) - 0.5 \times 1167 \times 0.035} = 1202 \text{ kg}$$

20) The example in 5) shews that with continuous blowing and the saltnesses here assumed of 3.5 % for the feed and 7 % for the boiler water, the weight blown off equals the weight of feed. During the 10 minutes we must therefore blow off 1167 kg of brine to equal the 1167 kg of feed evaporated. In consequence of the blowing only being performed periodically (i. e. every 10 minutes) the weight of brine to be removed is increased to 1202 kg, that is by $1202 - 1167 = 35$ kg, which proves that it is theoretically more economical to brine continuously than periodically. **Continuous blowing theoretically more correct than periodical.**

§ 34.

Weight of Feed.

Feeding from the sea.

1) If the feed is taken from *the sea*, the weight of feedwater *per hour* while the engines are at work is
   a) to replace the water evaporated, D kg (see Eq. 144, p. 278)
   b) „ „ „ „ blown off, W_{a_1} kg (see Eq. 149, p. 283)
   therefore together

$$W_s = D + W_{a_1} = D + D\frac{s}{s_x - s} = D\left(1 + \frac{s}{s_x - s}\right) \text{ kg} \quad \ldots (157)$$

Feeding from a jet condenser.

2) When feeding from a *jet condenser* s_c is to be substituted for s in this expression (see Eq. 150, p. 284).

Feeding from a surface-condenser.

3) With a *surface-condenser* and supplementary feed from *the sea* s_0 is to be put for s in Eq. 157 (see Eq. 152, p. 284).

Fresh supplementary feed.

4) It is only in the case of a *surface condenser* with fresh supplementary feed (from tanks, evaporators, &c) that the weight of feed water *per hour* equals the weight of steam used by the engine

$$Ws = D \text{ kg (see Eq. 144, p. 278)} \quad \ldots\ldots\ldots (158)$$

§ 35.

Coal-consumption.

Consumption classified.

1) **I. The total consumption.** Except in the case last referred to, of exclusively fresh feed, the coal-consumption K of a marine engine is made up of
   a) the weight of coal K_n producing the heat necessary for evaporating the water,
   b) the weight of coal K_v producing the heat which is lost by blowing off.

Useful coal-consumption.

2) *The useful coal-consumption* K_n is calculated as follows: to convert 1 kg of water at t_1^0 into steam at t^0

$$\lambda - t_1 = 606.5 + 0.305\, t - t_1 \text{ T. U.}$$

are necessary by Eq. 25, p. 26. As 1 kg of coal is capable of giving off $e = 4000$ to 6000 T. U. to the boiler water by § 21, 61, p. 205, we require

$$\frac{\lambda - t_1}{e} = \frac{606.5 + 0.305\, t - t_1}{e} \text{ kg}$$

of coal to produce this quantity of heat. If D kg of water were to be evaporated per hour (see eq. 144, p. 278) we should have a useful coal-consumption of

$$K_n = \frac{D}{c}\left(606.5 + 0.305\, t - t_1\right) \text{ kg} \quad \ldots\ldots\ldots (159)$$

When feeding on the system described in § 34, 4 p. 288, the above useful coal-consumption is also *the total hourly coal-consumption* of the engine.

3) For the purpose of estimating the consumption of a given engine λ is usually regarded in practice as constant and $= 640$ T. U.; with exclusively fresh feed, this gives the (useful and) total coal-consumption per hour Values for estimating.

$$K_n = D\left(\frac{640 - t_1}{e}\right) \text{ kg} \quad \ldots\ldots\ldots\ldots \quad (159^a)$$

in which expression the following empirical values of e are generally substituted:

$e = 6000$ T. U. with new first class boilers and the best coals,
$e = 5500$ „ „ good clean boilers and good coals,
$e = 5000$ „ „ not quite clean boilers and average coals,
$e = 4500$ „ „ dirty boilers and middling coals,
$e = 4000$ „ „ bad boilers and bad coals,
$t_1 = 15^0$ when feeding from the sea,
$t_1 = 40^0$ „ „ „ „ condenser (hot-well),
$t_1 = 90^0$ „ „ „ a feed-heater.

4) The *loss of heat* due to blowing-off is occasioned by the unevaporated portion of the feed-water being heated up from t_1 the temperature of the feed, to t that of the boiler-water, before it is blown off. Every kg of ejected brine, the average specific heat of which may be taken at 0.82, therefore withdraws from the boiler Consumption due to losses of heat.

$$0.82\,(t - t_1)\ T.\ U.$$

and there are thus lost by Eq. 149

$$0.82\, W_{a1}\,(t - t_1)\ T.\ U. \text{ per hour,}$$

to produce which quantity of heat

$$K_v = \frac{0.82\, W_{a1}\,(t - t_1)}{e} \text{ kg} \quad \ldots\ldots\ldots \quad (160)$$

of coal must be expended. The coal wasted by blowing-off is about 11 to 13 % of the useful coal consumption when the weight of water blown off equals the weight evaporated, and is about 6 to 7 % when the weight blown off is half the weight evaporated. These values refer to jet-condensers, but with surface condensers when the quantity blown off is only 0.04 to 0.08 of that evaporated, as may be the case with low-pressure boilers that are not quite tight (see § 33, 10) the coal lost by blowing-off is only 0.6 to 1.3 %, while with good tight high-pressure boilers where the water blown off is from 0.013 to 0.025 of that evaporated, the loss of coal sinks to about 0.2 to 0.4 %, so that it can be neglected.

Total coal consumption.

5) The hourly coal consumption of a marine engine therefore amounts to

$$K = K_n + K_r \text{ kg}$$

$$K = \mathrm{D}\left(\frac{640 - t_1}{e}\right) + 0.82\, W_{a_1}\left(\frac{t - t_1}{e}\right) \text{ kg,}$$

and for example, when feeding from the sea

$$K = \mathrm{D}\left(\frac{640 - t_1}{e}\right) + \mathrm{D}\left(\frac{s}{s_x - s} \times \frac{0.82\,(t - t_1)}{e}\right) \text{ kg}$$

$$K = \frac{\mathrm{D}}{e}\left[(640 - t_1) + \frac{0.82\, s\,(t - t_1)}{s_x - s}\right] \text{ kg} \quad \ldots\ldots \quad (161)$$

This corresponds to a daily consumption of

$$K_t = 0.024\,\frac{\mathrm{D}}{e}\left[(640 - t_1) + \frac{0.82\, s\,(t - t_1)}{s_x - s}\right] \text{ Tons} \quad \ldots \quad (161^a)$$

When feeding from a condenser s_c or s_0 respectively is to be substituted in this expression. The temperature t of the boiler water is found from the absolute boiler pressure p_x by the steam table on page 28. By what is said in 4), the second term in the large bracket can be neglected for modern compounds with high pressures. As explained in 2) this term disappears for triples with exclusively fresh feed.

Comparison values.

6) **II. Consumption per *IHP*.** This quantity K_{IHP} (per hour) is the usual criterion of the economy of different types of marine engines. In judging of an engine by this standard it is regarded *with its boilers* as an inseparable whole and of course no strict comparison with another engine alone can apply. In the comparative trials of the British gun-boats*) "Goshawk" and "Swinger" it was found that in consequence of the good design of the latter's boilers her single-expansion engines shewed a smaller coal consumption per *IHP* per hour than the compound engines of the former whose boilers were not such efficient evaporators. It was only when the *steam consumptions* of the two engines were compared that the superior economy of the compound engine was brought out. The steam consumption is therefore preferable to the coal consumption as a means of comparing engines. Of course when both are known the efficiency of the boiler is determined also.

Determination of the coal consumption per *IHP* per hour.

7) The coal consumption per *IHP* per hour K_{IHP} is determined by dividing the hourly consumption as obtained by weighing during a trial trip by the mean indicated horse-power.

Mean coal consumption.

8) The mean coal consumption of ordinary marine engines in continuous work at sea and therefore not in a state of special preparation as for a trial-trip, comes out, with favourable cut-off ratios, about as follows

*) Transactions of the Inst. of Naval Archts. London 1875. P. 100.

0.70 to 0.75 kg	in the latest quadruples	at 14 to 15	atmos. working press.
0.75 „ 0.85 „	„ recent triples*)	„ 10 „ 12	„ „ „
0.80 „ 0.90 „	„ older „	„ 8 „ 10	„ „ „
0.90 „ 1.00 „	„ later compounds	„ 6 „ 7	„ „ „
1.00 „ 1.25 „	„ older „	„ 4 „ 5	„ „ „
1.25 „ 1.50 „	„ medium-pressure compounds	„ 3	„ „ „
1.50 „ 1.75 „	„ low-pressure engines	„ 2	„ „ „

having surface-condensers, jackets, and superheaters,

1.75 „ 2.00 „ „ low-pressure engines at 2 atmos. working pressure, having jet condensers and neither jackets nor superheaters,

2.00 „ 2.50 „ „ low-pressure engines at less than 2 atmos. working pressure having jet condensers and neither jackets nor superheaters.

9) In sea service as a rule only the coal consumption for 24 hours is calculated and this quantity is of importance in designing the bunkers. To facilitate the computation of it the following table of R. ZIESE'S**) is appended. If the *IHP* of an engine is not known, it can be easily estimated from the grate surface, as explanied in § 64. The consumption per *IHP* per hour for any particular type of engine can be taken from the data given above in 8) and for safety the larger number is to be chosen for each case, especially when determining the required capacity of bunkers. **Daily consumption.**

Example: an engine of 5500 *IHP* working at 0.7 kg per *IHP* per hour, consumes according to the table $84 + 8.4 + 0.84 = 93.24$ tons per 24 hours.

10) **III. The smallest coal consumption**. Many periodicals, particularly English and American, in describing and recommending new systems of engines or designs of boilers, frequently report such small consumptions as having been obtained on trials that they sometimes lead to the inference that the limits of what is theoretically possible have been exceeded. Such statements injure respectable firms and often mislead shipping circles. A very simple process is therefore given below by which the smallest consumption theoretically attainable may be determined for any particular case. **Least coal consumption.**

*) Compare HAACK and BUSLEY. Die technische Entwickelung des Norddeutschen Lloyds und der Hamburger Packetfahrt-Gesellschaft. Berlin 1893. P. 201, where the average consumption of 39 triples in these companies' fleets are given.

**) R. A. ZIESE. Die Dreifach-Expansionsmaschine. Protokolle des St. Petersburger Polytechnischen Vereins. No. 69. 1886. P. 12.

Table of daily consumption in Tons.

IHP	kg of coals per *IHP* per hour																	
	0,4	0,45	0,5	0,55	0,6	0,65	0,7	0,75	0,8	0,85	0,9	0,95	1	1,1	1,2	1,3	1,4	1,5
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
10000	96,0	108,0	120,0	132,0	144,0	156,0	168,0	180,0	192,0	204,0	216,0	228,0	240,0	64,0	278,0	312,0	336,0	360,0
9000	86,4	97,2	108,0	119,8	129,6	140,0	151,2	162,0	173,0	183,5	194,4	205,0	216,0	237,6	249,2	270,8	302,0	324,0
8000	76,8	86,4	96,0	104,5	115,2	124,0	134,4	144,0	153,0	159,0	162,8	167,3	192,0	211,0	230,4	249,6	269,0	288,0
7000	67,2	75,6	84,0	92,4	100,8	109,6	117,6	126,0	134,4	142,0	151,2	159,7	168,0	190,0	201,6	218,4	235,0	242,0
6000	57,6	64,8	72,0	79,2	86,4	90,8	100,8	108,0	115,0	122,5	129,6	137,2	144,0	160,0	172,8	187,2	201,0	216,0
5000	48,0	54,0	60,0	66,0	72,0	78,0	84,0	90,0	96,0	102,0	108,0	114,0	120,0	132,0	144,0	156,0	168,0	180,0
4000	38,4	43,2	48,0	52,8	57,6	62,0	67,2	71,6	76,8	82,0	86,4	91,0	96,0	105.5	114,2	124,8	134,0	144,0
3000	28,8	32,4	36,0	40,6	43,2	46,3	50,4	54,0	57,6	60,5	64,8	68,6	72,0	79,2	86,4	93,6	101,0	108,0
2000	19,2	21,6	24,0	26,4	28,8	31,4	33,6	36,0	38.4	40,5	43,2	45,6	48,0	52,8	57,6	62,4	67,0	72,0
1000	9,6	10.8	12,0	13,2	14,4	15,8	16,8	18,0	19,2	20,4	21,6	22,8	24,0	26,4	28,8	31,2	33.5	36,0
900	8,6	9,72	10,8	12,0	13,0	14,0	15,0	16,1	17,2	18,3	19.4	20,5	21,6	23,7	25,8	27,0	30,0	32,4
800	7.7	8,64	9,6	10,5	11,5	12,4	13.4	14.4	15,4	15,9	17,3	16,7	19,2	21,1	23,1	25,0	26,0	28,8
700	6,7	7,56	8,4	9,2	10,0	10,8	11,7	12,6	13,4	14,2	15,1	16,0	16,8	19,0	20,1	22,0	23,5	24,2
600	5,8	6,48	7,2	7,9	8,6	9,6	10,0	10,8	11,6	12,2	12,9	13,7	14,4	16,0	17,2	18,7	20,0	21,6
500	4,8	5,40	6,0	6,6	7,2	7,8	8,4	9,1	9,6	10,2	10,8	11,4	12,0	13,2	14,4	15,6	16,8	18,0
400	3,8	4,32	4,8	5,3	5,8	6,3	6,7	7,1	7,6	8,2	8,6	9,1	9,6	10,5	11.4	12,5	13,4	14,4
300	2,9	3,24	3,6	4,0	4,3	4.7	5,0	5,4	5,8	6,0	6.5	6,9	7,2	7.9	8.6	9,4	10,0	10,5
200	1,9	2,16	2,4	2,6	2,9	3.15	3.3	3,6	3,8	4,0	4.3	4,6	4,8	5.3	5.7	6,2	6.7	7,2
100	0,96	1,08	1,2	1,3	1,44	1,58	1,7	1,8	1,9	2,0	2,20	2,3	2,4	2,6	2,9	3,1	3.3	3,6
90	0,86	0,97	1,08	1,2	1,30	1,40	1,5	1,6	1,7	1,8	1,94	2,0	2,2	2,3	2,6	2,7	3,0	3,2
80	0,77	0,86	0,96	1,0	1,15	1,24	1,3	1,44	1,54	1,6	1,73	1,69	1,9	2,1	2,3	2,5	2,6	2,9
70	0,67	0,76	0,84	0,9	1,00	1,09	1,15	1,26	1,34	1,42	1,51	1,60	1,7	1,9	2,01	2,2	2,35	2,42
60	0,58	0,65	0,72	0,8	0,86	0,96	1,00	1,08	1,16	1,24	1,30	1,37	1,44	1,6	1,73	1,9	2,00	2,16
50	0,48	0,54	0,6	0,66	0,72	0,78	0,84	0,92	1,06	1,07	1,08	1,14	1,20	1,32	1,44	1,6	1,68	1,80
40	0,38	0,43	0,48	0,53	0,60	0,65	0,64	0,72	0,76	0,80	0,86	0,92	0,96	1,05	1,14	1,25	1,34	1,44
30	0,29	0.33	0,36	0,40	0,43	0,47	0,50	0,54	0,58	0,61	0,65	0,69	0,72	0,79	0,86	0,94	1,00	1,08
20	0,19	0,22	0,24	0,26	0,30	0,314	0,33	0,36	0,38	0,41	0,43	0,46	0,48	0,53	0,57	0,62	0,67	0,72
10	0,10	0,11	0,12	0,13	0,144	0,158	0,17	0,18	0,19	0,20	0,22	0,23	0,24	0,26	0,29	0,31	0,33	0,36

Work of a perfect engine.

11) By Eq. 21, p. 19 the heat converted into work by 1 kg of a perfect gas in a CARNOT'S cycle is

$$A L = \frac{Q}{T}(T - T_1).$$

Similarly, in a perfect engine the heat converted into work by 1 kg of steam is

$$A L = \frac{r}{T}(T - T_1)$$

where r = the heat of evaporation, see Col. 8 of the steam table p. 28,

T = the absolute temperature of the steam at the beginning of the process, see Eq. 9 p. 9,

T_1 = the absolute temperature of the steam at the end of the process.

The Value of $\frac{r}{T}$ is

1.0006 at a working pressure of 13,25 atmos.

1.0454 at a working pressure of 10 atmos.
1.1488 „ „ „ „ „ 5 „
so that it may be taken as equal to 1 for present ordinary pressures. The difference of the *absolute* temperatures in the bracket is the same thing as the difference of the temperatures. We may therefore put

$$A L = t - t_1$$

in which t is the temperature of the admission steam,
and t_1 „ „ „ „ „ exhaust „
say 60^0 for an ordinary back-pressure of 0.2 atmos. The work of 1 kg of steam in a perfect engine is therefore approximately

$$L = \frac{t - t_1}{A} = 424\,(t - t_1) = 424\,(t - 60) = 424\,t - 25440 \text{ mk.}$$

12) One *IHP* corresponds to 75 mk per second or 270000 mk per hour and as 1 kg of steam can produce $424\,t - 25440$ mk of work, then one *HP* per hour will require Coal-consumption of a perfect engine.

$$\frac{270000}{424\,t - 25440} \text{ kg of steam.}$$

The useful evaporative power of 1 kg of coals is, by Eq. 97, p. 168 e_{v_n} kg of water, so that to evaporate the above weight of steam requires

$$\frac{270000}{e_{v_n}(424\,t - 25440)} \text{ kg of coal.}$$

The values of e_{v_n} to be substituted here are given in § 22, 2 p. 207 for different coals and various kinds of boilers. For other fuels they will be found in col. 10 of the table on p. 182.

13) By the preceding the steam consumption per *IHP* per hour of a quadruple expansion engine with a working pressure of 14 atmos. and therefore $t = 197^0$ is approximately calculated thus Example.

$$\frac{27000}{424 \times 197 - 25440} = 4.64 \text{ kg of steam}$$

Accordingly as we assume the evaporation ratio to be 8, 9, or 10, we get the smallest theoretical coal-consumption per *IHP* per hour at 0.58, 0.51 or 0.46 kg. In reality the engine will not do more than 75 $^0/_0$ of the theoretical work, so that the above quantities will be increased by $^1/_3$ to 0.77, 0.68, and 0.61 kg.

14) By calculating in this way the approximate smallest coal-consumption per *IHP* per hour from the boiler pressure, a criterion is obtained by which the credibility of any alleged consumption may be judged. If this is less than the figure obtained by calculation it is untrue — if it approaches it at all closely it must be received with distrust. Where greater accuracy is Usefulness of this calculation.

desired, the theoretical steam consumption can be determined by Eq. 137 p. 266 and this quantity divided by e_{v_n} gives the smallest coal-consumption per *IHP* per hour.

§ 36.

Numerical Relations between Engine Power, Coal-consumption, and Speed of Ship.

Law.

1) **I. Law of corresponding speeds.** *A ship, whose lineal dimensions are n times as great as those of its model, the resistances of which at the speeds v_1, v_2, v_3 are found to be R_1, R_2, R_3, will have the resistances $n^3 R_1$, $n^3 R_2$, $n^3 R_3$ at the speeds $v_1\sqrt{n}$, $v_2\sqrt{n}$, $v_3\sqrt{n}$.*

Correctness of the law.

2) After the first application of this law to the determination of the resistances of ships by W. FROUDE in 1870*), HELMHOLZ in 1873**) shewed that for ideal frictionless fluids it could be deduced from the general fundamental equations of hydrodynamics. In 1874 FROUDE, in his report on the towing experiments with the British corvette "Greyhound"***) and her model, admitted that the law can be correct for ships only when their total resistance is made up of skin-friction resistance, wave-making resistance, and eddy resistance and when the resistance is proportional to the surface against which it acts and to the square of the speed. RIEHN†) subsequently pointed out that even if the law were true under the above limitations it could still only be regarded us an empirical one of casual applicability, because it assumes the coefficients of skin friction of the ship and the model to be equal. Nevertheless it is used in practice for predicting speeds and gives results which are very useful. It is therefore applied in the following pages.

Ratio of the lineal dimensions.

3) Assuming that the resistance of a model moving through the water is proportional to the square of its speed and to the area of its immersed midship section $\mathfrak{M}$, then its resistance at any velocity v is

$$R = k\,\mathfrak{M}\,v^2$$

where k is a coefficient of resistance depending upon the special circumstances. At the *corresponding* speed the resistance of the ship would be

$$n^3 R = k\,\mathfrak{M}_1\,(v\sqrt{n})^2$$

*) Transactions of the institution of naval architects. London 1870. P. 88.

**) Monatsbericht der Königlichen Akademie der Wissenschaften zu Berlin. June 1873.

***) Transactions of the institution of naval architects. London 1874. P. 52.

†) Zeitschrift des Vereines deutscher Ingenieure. 1887. P. 9.

on the other hand the ship's resistance at the *same* speed as the model's is

$$R_1 = k\,\mathfrak{M}_1\,v^2;$$

hence we get

$$\frac{n^3 R}{R_1} = \frac{k\,\mathfrak{M}_1\,(v\sqrt{n})^2}{k\,\mathfrak{M}_1\,v^2} = n$$

$$R_1 : R = n^2 : 1 \quad \ldots\ldots\ldots\ldots \quad (162)$$

i. e. *At equal speeds the resistance of a ship is to that of its model as the square of the ratio of their linial dimensions (scale) is to unity.*

4) Calling the displacement of the ship T_1, we have $\sqrt[3]{T_1}$ as the side of a cube whose contents are equal to the displacement. If $\sqrt[3]{T}$ is the side of the cube equal to the model's displacement, then Ratio of Displacement.

$$n = \frac{\sqrt[3]{T_1}}{\sqrt[3]{T}}$$

and

$$\frac{R_1}{R} = \frac{n^2}{1} = \left(\frac{\sqrt[3]{T_1}}{\sqrt[3]{T}}\right)^2 = \frac{T_1^{2/3}}{T^{2/3}} \quad \ldots\ldots\ldots \quad (163)$$

i. e. *At equal speeds the resistance of a ship is to that of its model as the two-thirds powers of their displacements.*

5) These relations between a ship and its model are approximately true of two ships of similar form. Relations between similar ships.

6) As $R_1 = k\,\mathfrak{M}_1\,v^2$ is the resistance of a ship at speed v, the effective work done in propelling the ship at that speed is Work due to resistance at different speeds of ship.

$$R_1 \times v = k\,\mathfrak{M}_1\,v^2 \times v = k\,\mathfrak{M}_1\,v^3$$

and at another speed v_1

$$R_1'\,v_1 = k\,\mathfrak{M}_1\,v_1^{\,3}$$

assuming k to remain the same at both speeds. It then follows that

$$\frac{R_1\,v}{R_1'\,v_1} = \frac{v^3}{v_1^{\,3}}.$$

7) The work due to the resistance of the ship $R_1\,v$ expressed in horse-power is by § 30, 15 p. 253 the effective horse-power of the engine, and as we may assume the effective horse-power to be proportional to the indicated for speeds not too greatly different, we get Engine-power at different speeds of ship.

$$\frac{IHP}{IHP_1} = \frac{v^3}{v_1^{\,3}} \quad \ldots\ldots\ldots\ldots\ldots \quad (164)$$

i. e. *For the same ship the horse power is proportional to the speed cubed.*

Coal-consumption in equal times for different speeds of a ship.

8) At any ratio of cut-off greater than the most economical one, the consumption per *IHP* per hour increases *with the horse power.* At any smaller ratio of cut-off than the most economical one, the *IHP* diminishes while the hourly consumption per *IHP* increases. Nevertheless for cut-offs which do not differ too greatly from the most economical we may in general assume the consumption during a given time to be proportional to the *IHP* for the same engine

$$\frac{IHP}{IHP_1} = \frac{K}{K_1} \quad \ldots \ldots \ldots \ldots \ldots \quad (165)$$

And by Eq. 164 and 165

$$\frac{K}{K_1} = \frac{v^3}{v_1^3} \quad \ldots \ldots \ldots \ldots \ldots \quad (166)$$

i. e. *for the same ship and equal times the consumption varies as the speed cubed.*

Engine power and coal-consumption of similarly formed ships at the same speed.

9) Since for the same ship the horse-powers vary as the resistances and these latter — for similarly-formed ships at the same speed — are proportional to the two-thirds powers of the displacements, we have

$$\frac{IHP}{IHP_1} = \frac{K}{K_1} = \frac{T^{2/3}}{T_1^{2/3}} \quad \ldots \ldots \ldots \ldots \quad (167)$$

i. e. *with similarly formed ships at the same speed the horse-powers and consumptions vary as the two-thirds powers of the displacements.* The power required for equal speeds in similar ships increases in a lesser ratio than the displacement. It may in general be inferred from Eq. 167 that the coal consumption corresponding to cargoes of different magnitude increases more slowly than in direct proportion to them, or in other words that it is cheaper to carry large quantities than small ones. Hence it follows that large steamers must always be the cheapest to work provided they can get full cargoes and good despatch in loading and discharging.

Consumption for the same distance at different speeds.

10) Calling ΣK a ship's consumption for a run of m miles performed in l days at v knots, the hourly consumption is

$$K = \frac{\Sigma K}{24\, l}$$

and the speed

$$v = \frac{m}{24\, l}$$

If the ship makes the same run at another speed v_1 in l_1 days at a consumption ΣK_1, the hourly consumption is

$$K_1 = \frac{\Sigma K_1}{24\, l_1}$$

and the speed

$$v_1 = \frac{m}{24\, l_1}.$$

As by Eq. 166 $\quad \frac{K}{K_1} = \frac{v^3}{v_1^{\,3}}$

we must have

$$\frac{\frac{\Sigma K}{24\, l}}{\frac{\Sigma K_1}{24\, l_1}} = \frac{\frac{m}{24\, l} v^2}{\frac{m}{24\, l_1} v_1^{\,2}}$$

$$\frac{\Sigma K}{\Sigma K_1} = \frac{v^2}{v_1^{\,2}} \quad \ldots\ldots\ldots\ldots\ldots \quad (168)$$

i. e. *the same ship's consumption for equal distances varies as the squares of the speeds.*

11) **II. Examples.** The following examples illustrate the relations developed above and shew their use in estimating. **Examples of the practical application of these ratios.**

I. A steamer's engines indicate 1200 IP at 10 knots; how much must they indicate to drive her 12 knots?

$$\frac{IP}{IP_1} = \frac{v^3}{v_1^{\,3}};\ IP_1 = \frac{v_1^{\,3}\, IP}{v^3}$$

$$IP_1 = \frac{12^3 \times 1200}{10^3} = 2073{,}6\ IP.$$

II. A steamer burns 40 tons per day at an average speed of 9 knots; what will be her daily consumption at 11 knots?

$$\frac{K}{K_1} = \frac{v^3}{v_1^{\,3}};\ K_1 = \frac{v_1^{\,3} K}{v^3}$$

$$K_1 = \frac{11^3 \times 40}{9^3} = 73 \text{ tons.}$$

III. A steamer's consumption on a run from Hamburg to New York is 600 tons at an average of 11 knots. How much will she burn if the speed is reduced to 9 knots?

$$\frac{\Sigma K}{\Sigma K_1} = \frac{v^2}{v_1^{\,2}};\ \Sigma K_1 = \frac{v_1^{\,2}\, \Sigma K}{v^2}$$

$$\Sigma K_1 = \frac{9^2 \times 600}{11^2} = 402 \text{ Tons.}$$

IV. A steamer's bunkers contain 400 tons and they are emptied on a 12 knot run of 2500 miles. She is re-bunkered and makes a further run of 1900 miles at 11 knots. How many tons will be left in her bunkers on arrival?

On the first run she burns $\frac{400}{2500}$ tons per mile and on the second $\frac{x}{1900}$ tons. For *equal distances* the consumptions are to each other as

$$\frac{\Sigma K}{\Sigma K_1} = \frac{v^2}{v_1^2};\ \Sigma K_1 = \frac{v_1^2 \Sigma K}{v^2}$$

$$\frac{\frac{400}{2500}}{\frac{x}{1900}} = \frac{12^2}{11^2};\ x = \frac{400 \times 11^2 \times 1900}{2500 \times 12^2} = 255 \text{ Tons.}$$

therefore there are left in the Bunkers

$$400 - 255 = 145 \text{ Tons.}$$

V. A steamer's daily consumption is 60 tons at 13 knots average. She is delayed for several days by heavy weather and on the storm moderating it is found that only 80 tons are left in the bunkers. The nearest port is 800 miles off. To what speed must she be reduced in order to reach this port with her stock of coal?

Her consumption per mile at 13 knots was $\frac{60}{13 \times 24}$ and this must now be reduced to $\frac{80}{800}$.

These consumptions for *equal distances* are to each other as

$$\frac{\Sigma K}{\Sigma K_1} = \frac{v^2}{v_1^2};\ v_1^2 = \frac{v^2 \Sigma K_1}{\Sigma K}$$

$$\frac{\frac{60}{13 \times 24}}{\frac{80}{800}} = \frac{13^2}{v_1^2};\ v_1 = \sqrt{\frac{13^2 \times 80 \times 13 \times 24}{800 \times 60}}$$

$$v_1 = 9{,}3 \text{ Knots.}$$

VI. How many miles can a steamer travel at 10 knots if her bunkers contain 700 tons and her daily consumption at 15 knots is 100 tons?

Calling the daily consumption at 10 knots K_1, the consumptions in *equal times* are to each other as

$$\frac{K}{K_1} = \frac{v^3}{v_1^3};\ K_1 = \frac{v_1^3 K}{v^3}$$

$$\frac{100}{K_1} = \frac{15^3}{10^3};\ K_1 = \frac{10^3 \times 100}{15^3} = 29.6 \text{ tons}$$

As she only burns 29.6 tons at 10 knots, 700 tons will last her $\frac{700}{29.6}$ days and carry her $\frac{700 \times 10 \times 24}{29.6} = 5675$ miles.

VII. A steamer can travel 2000 miles at 12 Knots on 300 tons of coals. How many miles can she travel at 10 knots on the same coals and how many days will she take on the passage?

Her day's work at 12 knots is of course 288, so that her time for 2000 miles is $\frac{2000}{288}$ and her daily consumption

$$\frac{300}{\frac{2000}{288}} \text{ tons.}$$

Calling her daily consumption at 10 knots K_1 tons, we get, for *equal times*

$$\frac{K}{K_1} = \frac{v^3}{v_1^3};\ K_1 = \frac{v_1^3 K}{v^3}$$

$$\frac{\frac{\frac{300}{2000}}{12 \times 24}}{K_1} = \frac{12^3}{10^3};\ K_1 = \frac{300 \times 12 \times 24 \times 10^3}{2000 \times 12^3} = 25 \text{ tons.}$$

As her daily consumption is 25 tons for 10 knots she can steam $\frac{300}{25} = 12$ days and travel $12 \times 10 \times 24 = 2880$ miles. This question can also be solved without knowing the contents of the bunkers; thus

if ΣK is the total consumption for 2000 miles at 12 knots
and ΣK_1 „ „ „ „ „ „ „ „ 10 „
we get

$$\frac{\Sigma K}{\Sigma K_1} = \frac{v^2}{v_1^2};\ \Sigma K_1 = \frac{\Sigma K v_1^2}{v^2}$$

$$\Sigma K_1 = \frac{\Sigma K \times 10^2}{12^2} = 0.694\, \Sigma K$$

i. e. if the boat steams 2000 miles at 10 knots she consumes 0.694 of the contents of her bunkers, so that she can travel $\frac{2000}{0.694} = 2880$ miles if she goes on till they are empty.

VIII. A steamer can travel 3000 miles at 12 knots with 500 tons bunker capacity. Aftershipping 100 tons more outside the bunkers, she has to make a voyage of 4800 miles. What must be her average speed if her coals are to last for the run?

In the first case she burns $\frac{500}{3000}$ tons per mile, in the second she can consume $\frac{600}{4800}$. These consumptions for *equal distances* are to each other as

$$\frac{\Sigma K}{\Sigma K_1} = \frac{v^2}{v_1^2};\ v_1 = \sqrt{\frac{v^2\, \Sigma K_1}{\Sigma K}}$$

$$\frac{\frac{500}{3000}}{\frac{600}{4800}} = \frac{12^2}{v_1^2};\ v_1 = \sqrt{\frac{12 \times 600 \times 3000}{4800 \times 500}}$$

$$v_1 = 10.3 \text{ knots.}$$

IX. A steamer can travel 4000 miles at 12 Knots with a bunker capacity of 550 tons. She has to make a voyage of 3000 miles at her maximum speed of 15 knots. How much coal must she take on board besides filling her bunkers?

At 12 knots she burns $\frac{550}{4000}$ tons per mile, at 15 knots $\frac{x}{3000}$. These consumptions are to each other for *equal distances* as

$$\frac{\Sigma K}{\Sigma K_1} = \frac{v^2}{v_1^2};\ \Sigma K_1 = \frac{v_1^2\, \Sigma K}{v^2}$$

$$\frac{\frac{550}{4000}}{\frac{x}{3000}} = \frac{12^2}{15^2};\ x = \frac{15^2 \times 550 \times 3000}{4000 \times 12^2} = 644.5 \text{ tons.}$$

So that she must carry, besides filling her bunkers

$$644.5 - 550 = 94.5 \text{ tons.}$$

X. A steamer's engines consume on an average 1.2 kg of coal per *IHP* per hour when indicating 2500, her speed being 14 knots. What is the greatest power she must indicate in order to make a passage of 2400 miles with her bunker capacity of 350 tons?

At 14 knots she burns per mile $\frac{1.2 \times 2500}{1000 \times 14}$ tons; on the proposed run she must only burn $\frac{350}{2400}$. The consumptions for *equal distances* are to each other as

$$\frac{\Sigma K}{\Sigma K_1} = \frac{v^2}{v_1^2};\ v_1 = \sqrt{\frac{v^2\, \Sigma K_1}{\Sigma K}}$$

$$\frac{\frac{1,2 \times 2500}{1000 \times 14}}{\frac{350}{2400}} = \frac{14^2}{v_1^2};\ v_1 = \sqrt{\frac{14^2 \times 350 \times 1000 \times 14}{2400 \times 1,2 \times 2500}}$$

$$v_1 = 11.5 \text{ knots.}$$

So that her speed for the proposed run must be 11.5 knots. As she requires 2500 *IHP* for 14 knots, the *IHP* for 11 knots will be

$$\frac{IHP}{IHP_1} = \frac{v^3}{v_1^{\,3}};\ IHP_1 = \frac{v_1^{\,3}\, IHP}{v^3}$$

$$\frac{2500}{IHP_1} = \frac{14^3}{11{,}5^3};\ IHP_1 = \frac{11.5^3 \times 2500}{14^3}$$

$$IHP_1 = 1385.5\ IHP.$$

Sixth Division.

Estimation of the power required for proposed steamers.

§ 37.

Resistance of ships.

Determination of a ship's resistance.

1) **I. Determination of a ship's resistance.** For more than a century attempts have been made to devise a formula based upon data obtained from repeated experiments, by means of which the resistance of a proposed ship should be calculable beforehand. Unfortunately this object has not been achieved, as no formula exists which gives unquestionably accurate results for all types of ships and all systems of propulsion. The calculation of the required engine power for a certain ship at any speed, which would at once follow from the resistance if this were known accurately beforehand, is therefore always as yet performed by means of more or less approximate formulæ, of which KIRK'S is perhaps the one most used in practice and is given in § 39, 13. The most notable of the earlier and recent experiments and formulæ relating to this subject are recorded below in order to afford a clear view of the methods adopted in ascertaining the resistance of ships and of the progress which has hitherto been made in the investigation of the question.

Methods of determining the ships resistance.

2) Most formulæ for calculating the resistance of a ship are based upon experiments carried out with a limited number of steamers under more or less special conditions. It is only in recent years that designers have begun to adopt the elder FROUDE's system of predicting the resistance of a proposed ship from that of its model by means of the law of corresponding speeds (§ 36, 1 p. 294) and subsequently to test the correctness of the calculation by progressive trials of the ship, which were first instituted by DENNY. This method is at present regarded as the best and will be fully described further on in connection with TIDEMANN's experiments, and KIRK's formula (§ 39, 13) which is based upon it is therefore

the one chiefly employed in calculating the engine power for proposed steamers.

3) Such attempts as have hitherto been made to ascertain the resistance of steamers were founded on **Different kinds of trials.**
   a) steaming trials,
   b) stopping „ ,
   c) towing „ .

4) In steaming trials of screw steamers a dynamometer is put in place of the thrust-block so as to register directly the thrust delivered by the propeller to the ship. If at the same time a rotating dynamometer is applied between the thrust-block and the engines, measuring the useful horse-power of the latter, we get all the particulars of the ship's resistance with an accuracy and completeness unattainable by any other method. Trials of this sort mostly fail however, in consequence of the difficulty of constructing such dynamometers for large engines and of the untrustworthiness of the results they give for small ones. GUÈDE and JAY's trials of the French despatch boat "ELORN" in 1861 (§ 38, 16) and ISHERWOOD's trials of a steam launch belonging to the U. S. Navy in 1870 (§ 38, 36) were carried ont by this method. **Steaming trials.**

5) b. *Stopping trials* consist in steaming ahead at full power, suddenly stopping the engines, and observing the subsequent extinction of speed. This is done on a measured mile buoyed off at certain distances. By observing the time-intervals at which the buoys are successively passed, plotting them graphically, and taking into account the vis viva of the ship at the different speeds, a curve of resistance can be constructed. But this method also has its defects, viz. that the ship's resistance is greatly altered when the propeller is no longer working and that the result is very much affected by the drag of the motionless propeller itself. In spite of these drawbacks, stopping trials are often the only ones available, for want of the apparatus necessary for any other kind of trial. **Stopping trials.**

6) c. Towing trials are the simplest means of determining the resistance. The ship is towed at various speeds on a measured mile and the pull on the towrope measured by a dynamometer. This method has the advantage of being attended by very small errors of observation with good instruments. On the other hand it has this drawback that the resistance of the ship when towed is not the same as when it is driven by its own propeller, because the latter's action upon the water in the ship's run is absent, and secondly because the towed ship's resistance is affected by the "wash" of the tug. The former circumstance **Towing trials.**

is of great importance, as it has been shewn by the experiments of FROUDE, TIDEMANN, and ISHERWOOD, referred to further on, that with fast running single screws the augmentation of resistance due to the propeller's action amounts to from 20 to 45 % of the ship's resistance when towed with the propeller at rest. The resistance of the British sloop of war "Greyhound" was found from trials of this sort in 1871 by FROUDE. (See § 39, 15.)

Use of the trials.

7) If the resistance of a ship is carefully determined by all three methods in succession, the results will be found to differ somewhat from each other as may be inferred from the foregoing. But each set of trials by itself as well as the whole series will shew how the resistance of one and the same ship varies, *ceteris paribus*, with the speed and also how the resistance differs at the same speed for ships of various types. Thus the first basis of an algebraical expression for a ship's resistance is obtained.

Various Kinds of coefficients of performance.

8) **II. Coefficients of performance.** Before any such expression for the resistance existed, it was customary until lately, especially in the British Navy, to be satisfied with estimating the power of proposed steamships by comparison with similar ones already existing. For this purpose *coefficients of performance* or *Admiralty coefficients* were developed in the course of time. They are of three different kinds, destinguished by the constant (derived from the ship's dimensions) with which the quotient $\frac{v^3}{IHP}$ is multiplied, and are generally known as

a) C, the midship section coefficient,
b) C_1, the displacement „ ,
c) C_2, the surface „ .

English midship section coefficient.

9) a. **The midship section coefficient.** The ship's resistance was shewn in § 36, 3 p. 294 to be expressed by

$$R_1 = k\,\mathfrak{M}\,v^2$$

where $\mathfrak{M}$ is the area of immersed midship section and k the coefficient of resistance. The work done in overcoming this resistance is therefore

$$R_1\,v = k\,\mathfrak{M}\,v^3.$$

Assuming the coefficient k to be replaced by another k_1 which takes account of all losses of effect — from those relating to the steam in the cylinder down to those of the propeller, we can write

$$IHP = k_1\,\mathfrak{M}\,v^3$$

or

$$\frac{k_1\,\mathfrak{M}\,v^3}{IHP} = 1$$

$$\frac{1}{k_1} = C = \frac{v^3 \mathfrak{M}}{IHP} \quad \ldots \ldots \ldots \ldots \quad (169)$$

Upon the supposition that the frictional resistances of similar ships and engines are proportional to the IHP, this coefficient of resistance C is a criterion of the combined efficiency of the engine, propeller, and form of ship.

10) In the French Navy the midship section coefficient is derived in a somewhat different manner and designated by the letter M. The expression has been brought into this form, French midship section coefficient.

$$IHP = k_1 \, \mathfrak{M} \, v^3$$

$$\frac{IHP}{\mathfrak{M}} = k_1 v^3; \quad \sqrt[3]{\frac{IHP}{\mathfrak{M}}} = v \sqrt[3]{k_1}$$

$$\frac{v \sqrt[3]{k_1}}{\sqrt[3]{\frac{IHP}{\mathfrak{M}}}} = 1$$

$$\frac{1}{\sqrt[3]{k_1}} = M = \frac{v}{\sqrt[3]{\frac{IHP}{\mathfrak{M}}}} \quad \ldots \ldots \ldots \ldots \quad (170)$$

11) b. **The Displacement coefficient.** As two ships may have exactly equal areas of midship section with different waterlines, the coefficient C, *ceteris paribus*, would not shew the influence of full or fine form. In order to take this circumstance also into account, the displacement is regarded as a cube (see § 36, 4, p. 295), a side of which is $\sqrt[3]{T}$, and therefore its cross section $\left(\sqrt[3]{T}\right)^2 = T^{2/3}$. The area of this imaginary cross section is substituted for the area of midship section in Eq. 169 and gives the coefficient Displacement coefficient.

$$C_1 = \frac{v^3 \, T^{2/3}}{IHP} \quad \ldots \ldots \ldots \ldots \quad (171)$$

12) c. **The surface coefficient** was introduced very much later than the other two by RANKINE who called it the *propulsive coefficient*. In this the place of the area of midship section, or of $T^{2/3}$, is taken by the *augmented surface* of the ship, the signification of which is fully gone into in § 39, 3. Surface coefficient.

The expression then becomes

$$C_2 = \frac{v^3 \, S_1}{IHP} \quad \ldots \ldots \ldots \ldots \quad (172)$$

13) In Germany and in England, the displacement coefficient C_1 is most used in comparative calculations; in France however the midship section coefficient M is almost exclusively employed. The approximate accuracy of these coefficients follows from the fact that the resistance of a ship essentially depends on Accuracy of the coefficients.

Table of Coefficients of Performance of 52 modern Steamers.

Kind of ship	Name	Flag	Year of trial	Built by	Built at	Length L in m	Breadth B in m	Approxto $\frac{L}{B}$	Immersed $\boxtimes$ in sqm.	Displacement T in Tons	IP	Speed v	Midship section coefficient C	Displacement-coefficient C_1	French coefficient M	$\frac{IP}{\boxtimes}$	$\frac{IP}{T}$
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
				I. War-ships.													
Large twinscrew iron-clads	Wörth	German	1894	Germania	Kiel	108.00	19.5	5.5	128.00	10040	10289	17.23	684.96	231.37	3.992	80.38	1.024
	Royal Sovereign	English	1892	Ship: Royal Dockyard engine: Humphrys	Portsmouth	115.80	22.8	5.0	162.40	14150	13363	17.50	701.14	234.63	4.023	82.28	0.944
	Marceau	French	1891	Compagnie des forges et chantiers	Marseilles	100.60	20.00	5.0	139.04	10581	11017	16.19	576.51	185.64	3.491	79.24	1.041
Smaller twinscrew iron-clads	Beowulf	German	1892	Weser	Bremen	73.00	14.92	4.9	70.60	3500	4866	15.06	533.48	161.81	3.673	68.92	1.390
	Monterey	American	1893	Union Iron Works	St. Francisco	78.02	17.98	4.3	71.72	4000	4987	13.60	389.41	127.10	4.112	69.53	1.247
	Furieux	French	1888	Govt. Dockyard	Cherbourg	72.55	17.72	4.1	104.00	5560	4930	13.92	598.53	171.70	3.846	47.40	0.887
Triple screw ships	Kaiserin Augusta	German	1893	Germania	Kiel	118.30	15.60	7.6	87.75	5937	10635	20.23	735.67	255.25	4.088	121.19	1.791
	Columbia	American	1894	Cramp & Sons	Philadelphia	125.45	17.73	7.1	104.00	7350	17991	22.80	740.80	249.60	4.092	173.00	2.448
	Dupuy de Lôme	French	1893	Govt. Dockyard	Brest	114.00	15.70	7.2	89.97	6297	14000	20.00	553.37	194.86	3.718	155.61	2.223
Large twinscrew ships	Hohenzollern	German	1893	Vulcan	Stettin	116.60	14.05	7.9	65.20	4180	9446	21.60	748.77	276.84	4.113	144.87	2.260
	Blake	English	1891	Ship: Royal Dockyard engine: Maudslay	Chatham	114.30	19.81	5.8	137.49	9000	14525	19.12	712.21	208.21	4.044	105.64	1.614
	Cécille	French	1890	Forges et chantiers de la Méditerranée	La Seyne	115.50	15.00	7.7	77.81	5766	10680	19.44	576.15	221.19	3.769	137.26	1.852
Smaller twinscrew ships	Falke	German	1891	Imperial Dockyard	Kiel	76.00	10.20	7.6	33.40	1545	2909	16.92	599.46	222.54	3.817	87.09	1.883
	Barracouta	English	1891	Ship: Royal Dockyard engine: Palmer	Sheerness	67.00	10.66	6.3	39.39	1580	2125	14.90	660.03	211.17	3.943	53.95	1.345
	Condor	French	1887	Govt. Dockyard	Rochefort	68.00	8.90	7.6	30.74	1272	3582	17.78	519.23	184.22	3.640	116.53	2.816
Twin screw torpedo-boats	Cushing	American	1890	Herreshoff	Bristol R. I.	42.00	4.34	9.7	4.43	105.3	1754	22.48	308.85	144.43	3.061	395.94	1.6660
	No. I to V	Brazilian	1893	Schichau	Elbing	46.50	4.90	9.5	4.63	138.0	2200	28.00	496.90	266.40	3.580	475.16	1.5960
	No. I and II	Japanese	1891	Schichau	Elbing	39.00	4.52	8.6	4.02	84.5	1150	23.10	463.80	206.40	3.500	236.07	1.3610
Single screw torpedo-boats	Sweaborg	Russian	1886	Normand	Havre	46.70	3.76	12.4	3.58	96.0	800	19.70	368.28	200.36	3.246	223.46	8.333
	No. 92	English	1894	Thornycroft	Chiswick	43.40	4.74	9.1	4.50	129.0	2600	24.52	274.65	144.76	2.944	577.78	20.155
	No. I	Chinese	1886	Schichau	Elbing	43.00	5.00	8.6	4.10	92.0	1500	24.60	438.00	202.26	3.439	365.85	16.304

Note: "Wörth", "Royal Sovereign" and "Marceau" are sea-going battle ships. "Beowulf", "Monterey", "Furieux" are coast defence ships. "Kaiserin Augusta", "Columbia" and "Dupuy de Lôme" are cruisers the last of which carries light side-armour. "Hohenzollern" is a despatch-boat, "Blake" and "Cécille" are large cruisers. [illegible] and "Barracouta" are ordinary small cruisers. "Condor" is a torpedo cruiser.

II. Merchant ships.

a) Screw steamers.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
Fast twinscrew steamers	Campania	English	1893	Fairfield	Glasgow	182.88	19.81	9.2	142.1	16000	26000	22.6	679.27	281.90	3.981	182.97	1.625
	Paris	American	1889	Thomson	Glasgow	160.72	19.26	8.3	135.2	13000	18350	21.0	736.84	279.0	4.086	135.72	1.412
	Normannia	German	1890	Fairfield	Glasgow	152.40	17.45	8.7	115.0	10500	16244	20.5	657.64	254.31	3.936	141.25	1.547
	Fürst Bismarck	„	1891	Vulcan	Stettin	153.20	17.54	8.7	107.7	10163	16412	20.7	626.53	253.52	3.875	152.39	1.615
Fast single screw steamers	Umbria	English	1884	Fairfield	Glasgow	152.40	17.37	8.8	112.6	10500	14321	20.1	687.29	271.90	3.997	127.18	1.364
	Spree	German	1890	Vulcan	Stettin	141.12	15.70	9.0	95.7	8925	12712	20.1	657.97	274.82	3.939	132.83	1.424
	Lahn	„	1887	Fairfield	Glasgow	136.55	14.88	9.2	96.3	7700	8929	19.0	796.29	299.54	4.198	92.72	1.160
Single screw mail steamers	Scandia	„	1889	Vulcan	Stettin	113.70	13.56	8.3	83.2	5700	3409	13.0	577.18	205.60	3.771	40.97	0.600
	Russia	„	1889	Laird	Birkenhead	113.70	13.64	8.3	86.2	5500	2725	12.0	588.40	197.58	3.795	31.61	0.496
	München	„	1889	Fairfield	Glasgow	118.87	14.17	8.4	87.0	6200	2695	12.5	678.70	244.59	3.980	30.98	0.434
Large cargo steamers	Helvetia	„	1889	Wallsend	Newcastle	96.30	12.20	8.0	68.6	3700	1459	10.7	620.02	200.86	3.862	21.27	0.394
	Cheruskia	„	1890	Stephenson	Hebburn	101.50	12.57	8.0	63.0	4300	1570	10.5	500.03	194.98	3.595	24.92	0.365
	Venetia	„	1891	Reiherstieg	Hamburg	97.53	12.14	8.0	65.2	3750	1540	11.0	606.59	208.62	3.834	23.62	0.411
Smaller cargo steamers	Reiher	„	1883	Earle	Hull	68.58	8.53	8.0	34.4	1200	700	10.5	612.37	186.75	3.846	20.35	0.583
	Schwan	„	1884	Tecklenborg	Geestemünde	68.58	9.14	7.5	41.0	1500	500	9.5	756.79	224.70	4.127	12.20	0.330
	Sumatra	„	1889	Howaldtswerke	Kiel	52.00	8.53	6.1	22.5	750	340	10.0	712.34	242.79	4.145	15.11	0.453
Screw tugs	Dahlström	„	1887	Howaldtswerke	Kiel	23.77	6.09	3.9	6.56	95	133	9.5	455.20	134.22	3.484	20.27	1.400
	Farrapo	„	1893	Howaldtswerke	Kiel	41.78	7.01	5.9	13.93	325	340	11.0	587.00	185.05	3.792	24.40	1.046
	Stuttgart	„	1894	Howaldtswerke	Kiel	26.84	5.69	4.7	8.31	130	258	11.0	461.47	132.39	3.500	31.05	1.985

b) Paddle steamers.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
Large paddle steamers	Marie Henriette	Belgian	1893	Cockerill	Seraing	103.63	11.58	9.0	29.50	1780	8134	22.20	427.07	197.56	3.411	275.73	4.570
	La Marguerite	English	1894	Fairfield Co.	Glasgow	100.58	12.19	8.3	32.00	1868	7500	22.00	489.03	215.34	3.568	234.37	4.015
	Mona's Queen	English	1886	Shipbuilding Co.	Barrow	97.53	11.58	8.9	30.09	1825	4858	18.90	450.13	207.51	3.471	161.45	2.662
Medium paddle steamers	Mary Powel	American	1867	Fletcher	New York	87.17	10.44	8.3	18.58	800	1490	17.20	683.0	293.7	3.988	80.19	1.862
	Kaiseradler	German	1878	Germania	Kiel	81.65	10.36	7.9	30.02	1620	2650	15.74	475.0	200.8	3.032	88.27	1.636
	Aegir	Danish	1880	Burmeister & Wain	Copenhagen	60.95	7.31	8.3	15.42	570	950	13.50	429.92	178.04	3.418	61.61	1.667
Smaller paddle steamers	Willkommen	German	1884	Tecklenborg	Geestemünde	54.00	7.96	6.8	11.71	449	481	10.80	330.12	153.57	3.130	41.07	1.071
	Hecht	„	1885	Weser	Bremen	54.86	6.10	9.0	6.90	275	600	13.00	271.95	165.54	2.934	86.95	2.182
	Kehrewieder	„	1889	Tecklenborg	Geestemünde	56.69	7.93	7.1	12.67	499	550	11.77	404.31	186.51	3.349	43.41	1.102
Paddle tugs	Hercules	„	1887	Tecklenborg	Geestemünde	41.45	7.08	5.8	12.73	363	374	11.25	521.68	193.73	3.646	29.38	1.030
	Centaur	„	1889	Weser	Bremen	40.00	6.00	6.6	7.10	220	350	10.50	252.78	120.53	2.862	49.30	1.600
	Bessarabetz	Russian	1891	Howaldtswerke	Kiel	49.98	7.01	7.0	10.00	332	603	12.75	369.99	164.80	3.251	60.30	1.816

the wetted surface and that this again, for ships of somewhat similar form is nearly proportional to the displacement. For steamers of usual proportions the wetted surface is also in a fairly constant ratio to the area of midship section. If T is the displacement, $\sqrt[3]{T}$ is the side of a cube of the same volume and $T^{2/3}$ is the surface of one of its faces, five of which would be immersed, so that the wetted surface of the cube would be $5\ T^{2/3}$.

Calculation of the coefficients.

14) In order to collate as many comparative data as possible, it is still customary in Germany to work out the coefficients C and C_1 in English measure, while the French always calculate their coefficient M on the metrical system. Accordingly in the table on the preceding page, which gives the coefficients of performance of 52 modern steamers

for coeffs. C and C_1	v	denotes	knots per hour,
	$\mathfrak{M}$	„	area of midship section in sq. ft engl.,
	T	„	displt. in tons engl.
for coeff. M	v	„	knots per hour,
	$\mathfrak{M}$	„	area of midship section in sq. metres,
	T	„	displt. in metrical tons = 1000 kg.

To convert sq. metres into sq. ft engl. multiply by 10.7642. German and English tons can be regarded as equal, the English $T^{2/3}$ being only about 1 % greater than the corresponding German value.

Various theories.

15) **III. Elements of a ship's resistance.** — Different views are held as to the causes which produce the resistance and as to the magnitude of their effects. But in general two principal theories may be distinguished which we shall call

a) the displacement theory,
b) the stream-line theory.

They are both described below, but without regarding the intermediate distinctions between them which have been made by some authors.

Displacement-theory.

16) a. **The displacement-theory** is the older one and regards the total resistance W of a ship floating in water of unlimited extent as generally divided into

α) form-resistance W_f,
β) friction „ W_r,
γ) air „ W_l,

and but rarely a fourth term

δ) wave-resistance W_w.

17) α. *The Form-resistance* W_f arises from the displacement or pushing aside of the hitherto motionless water which, in consequence of its inertia and cohesion, only gives way comparatively slowly and exerts a greater hydrostatic pressure against the ship's entrance and a smaller one against her run, where it is flowing together again by its own weight, than is exerted against these parts of the ship when she is floating motionless. From this it is evident that the form-resistance depends upon the form of the ship's immersed body and the speed at which she progresses. Ships with fine entrance and run will therefore experience less form-resistance than those with very full lines. The influence of the speed upon the form-resistance is usually assumed to be in proportion to the speed squared. MIDDENDORF and RIEHN who include the wave-resistance W_w in the form-resistance, regard the latter as proportional to the 2.5 th power of the speed, whereas THORNYCROFT takes the 1.7 th power. In general the form-resistance is the largest component of the total resistance. It is only in very fine and fast ships with very large surface in proportion to their displacement that the friction-resistance preponderates. Form-resistance.

18) β. *The friction-resistance* W_r arises from the friction of the water on the ship's skin and therefore primarily depends upon its quality and condition. Although friction in general is supposed to be independent of velocity, the friction of sea water on ships is almost universally regarded as varying with their speed v. DUPUY DE LÔME takes it as proportional to $\sqrt[3]{v}$, FROUDE from his experiments, to $v^{1.83}$, TREDGOLD, BOURGOIS and RANKINE to v^2, THORNYCROFT to $v^{3.7}$, and GUÈDE and JAY to v^4. Opinions as to the coefficients of friction are therefore equally at variance as those on form-resistance. As the friction-resistance for full ships at low speeds was formerly considered to be much smaller than the form-resistance, early writers either disregarded friction-resistance altogether or included it in the form-resistance and therefore looked upon the form-resistance only as the total. Friction-resistance.

19) γ. *The air-resistance* W_l to which the unimmersed parts of a ship are exposed is, like the water-resistance, also composed of various elements, which it is impossible to determine separately. In a calm the air-resistance is very small, so that it only affects the total resistance to an inappreciable extent and is probably never greater than the errors inherent in the calculation of the total resistance by any of the existing more or less approximate formulæ. As, besides, trial-trips nearly always take place in moderate weather, the air-resistance has usually been neglected. Air-resistance.

Nevertheless it is advisable to estimate it for fast ships of unusual form or with high deck erections. BOURGOIS and FROUDE have attemptend to obtain numerical values for the air-resistance. In most of the formulæ it is taken as proportional to the area of the emerged portion of the ship projected on the plane of the midship section, and to the speed squared.

Wave-resistance. 20) *δ. The wave-resistance* W_w was first taken notice of by BOURGOIS and represents that increment of the pure form-resistance occasioned by the heaping up of the water at the fore-part of the ship, — *the bow-wave*, and by a wave-trough which under certain circumstances is formed at the stern. If instead of this trough, there is a crest, — *the stern-wave*, the resistance is thereby diminished as explained below in 26). In general the wave-resistance varies as the "height due" $\frac{v^2}{2g}$ and therefore as the speed squared. It also depends upon the ship's form. In view of the difficulty of expressing the wave-resistance with mathematical exactness and as the consequent complication of the formula for total resistance detracts from its practical utility, it has become usual to include the wave-resistance in the form-resistance by the choice of suitable coefficients.

Stream-line theory. 21) b. **The stream-line theory,** which was developed in recent times by RANKINE and FROUDE, assumes the ship to be at rest and the water to flow past her at a certain velocity and split up into separate small strips or "streams". These streams are diverted laterally at the ship's fore-body and, according to RANKINE'S view, pass over her surface in lines of least distance or *stream-lines*. The diversion of the streams causes a change in their velocity and therefore absorbs a certain amount of energy.

At the stern the streams close up together again, giving out energy, which to some extent makes up for that absorbed at the bow. From this it appears that this deviation of the streams to let the ship through, which is supposed to occasion the form-resistance, cannot account for so large a fraction of the total resistance as might be expected and as is assumed by the displacement-theory. The total resistance W must therefore be made up of other components, which FROUDE calls

α) friction-resistance W_r,
β) eddy „ W_v,
γ) wave „ W_w,
δ) air „ W_l.

Friction-resistance. 22) *α. The friction-resistance* W_r, arising from the gliding of the water particles along the wetted surface of the ship, depends upon the area of this surface, its length, its condition

of smoothness or roughness, and has been found to vary besides as the 1.83 th power of the speed. It is however not appreciably influenced by the form and proportions of the ship unless she has a very unusual model and bad lines. At low speeds, of from 6 to 8 knots, the friction-resistance is 80 to 90 % of the total, and must be estimated at 50 to 60 % of the total for the fastest ship with clean bottom. This proportion is considerably increased when the bottom is foul.

23) *β. The eddy-resistance* W_v is generated by the friction between the ship and the water particles communicating a forward motion to the latter and forming eddies in the wake. But with well formed ships these eddies possess only a low velocity and energy so that the work they absorb is but an unimportant constituent of the total. It is assumed that the eddy-resistance is directly proportional to the friction-resistance and under ordinary circumstances about 8 to 10% of it, the eddy-resistance may however be appreciably increased by an unusually short and "hard" run. **Eddy-resistance.**

24) *γ. The wave-resistance* W_w is occasioned by the surface disturbance consequent upon the diversion of the streams at the bow. This diversion retards the motion of the streams, thus increasing the pressure and causing a rise of level — the bow-wave. In the middle of the ship's length the streams attain their greatest velocity, a reduction of pressure takes place and therefore a wave-trough is formed. At the after body the streams are again retarded and, as at the bow there is a rise of level, — the stern-wave. The resistance due to the formation of these waves depends chiefly upon the form and proportions of the ship. Its relation to the friction-resistance as well as its absolute magnitude vary with a number of circumstances of which the most important are the ship's form and the ratio of the length of entrance and run to her proposed full power speed. **Wave-resistance.**

25) SCOTT RUSSELL was the first, in his wave-line theory*), to call attention to the importance of the wave-resistance and its relation to the length of the entrance and run. From his researches SIR WILLIAM WHITE**) deduces the rule that for a maximum speed of v knots the length of the entrance should be 0.1714 v^2 metres, and that of the run 0.1144 v^2 metres, i. e. $^2/_3$ rds. of the length of entrance. Also that between the entrance and the run a parallel middle body may be inserted of any length we please, which will only increase the friction- **Ship's length according to Scott Russell.**

*) J. SCOTT RUSSELL. The modern system of naval architecture. Vol. I. Part. I. p. 225. London 1865.

**) W. H. WHITE. A manual of naval architecture. II. Edit. p. 453. London 1882.

resistance. It has since been established that the length of run is the more important of the two, as many ships with a shorter entrance than that given by the rule have not shewn a proportionately greater bow wave, whereas a shorter run has given bad results. For steamers of high speed, 20 knots and over, the rule gives too short a run and experience shews that this should be longer than the entrance if full advantage is to be taken of the engine power.

Ship's length according to Froude.

26) The elder FROUDE endeavoured to solve the question of a suitable length of ship by trials, and found that the wave-resistance is influenced by the total length as well as by that of the entrance and run, particularly at greater speeds than those suitable to these dimensions in each case. He established the fact that the position of the stern wave has a very sensible effect upon the resistance; if its crest is situated at about the middle of the run, it exerts a pressure in that region which is equivalent to a reduction of the resistance. But if a wave-trough is formed there the resistance is increased. The younger FROUDE has also investigated with greater exactness the effect of wave-resistance by means of towing experiments with models (see § 40, 6) and shewn that the length of parallel middle body, so far as it affects the position of the stern-wave, may either increase or diminish the resistance.

Unsuitable length of ship.

27) It thus appears that for every ship there is a limiting speed, above which a small increase of speed is attended by a disproportionate increase of resistance, and this particular speed is fixed by the length of entrance and run. If under certain circumstances it is necessary to drive a ship at a speed greater than that at which the waves generated at her bow and stern naturally travel, her ratio of IHP to displacement becomes extraordinarily great compared with that of the fastest and most powerful sea-going ships. For instance, torpedo-boats of 20 to 28 knots speed and about 40 to 50 metres long would require by the above rules a length of entrance and run of 70 and 130 m. respectively, in order that the usual ratio of IHP to displacement should not be exceeded. Whereas in the largest and fastest cruisers this ratio of $\frac{IHP}{T}$ lies between 1.5 and 2.5 (see the table p. 306), it rises for these torpedo-boats as high as from 15 to 20.

Air-resistance.

28) *δ. The air-resistance* W_l is also usually neglected in the stream-line theory on account of its generally small amount. When taken account of, however it is based upon an experiment of FROUDE'S, according to which it = 0.0017 ℔s engl. per sq.

ft. engl. of plane surface at a speed of 1 ft. engl. per second. Assuming this resistance to vary as the square of the speed, it would equal 1 ℔ per sq. ft. engl. = 4.8826 kg per sq. m. at 15 knots. It may here be remarked that D'AUBUISSON*) found an air-resistance of 6.46 kg per sq. m. at a velocity of 7 m. per second and that the wind pressure given by WEISBACH'S**) formula for the same velocity is also rather over 6 kg per sq. m., so that further trials must determine the trustworthiness of FROUDE'S data.

Friction resistance the most important

29) As appears from the foregoing, friction-resistance, under ordinary circumstances, forms the greatest part of the total resistance, the eddy resistance is very small, and the wave-resistance for well-formed ships is inconsiderable. Disregarding the air-resistance also, which in any case is of very uncertain amount, we can, with RANKINE and KIRK, express the total resistance by the friction-resistance alone if we choose suitable coefficients of friction, in order to get a simple practically applicable formula such as is now mostly used.

Designations.

30) The following designations are used throughout the formulæ on the succeeding pages, so that a comparison may be made between the different authors' views; viz.

L length of the ship on the water-line,

L_1 „ „ „ forebody from ford. perp. to midships,

L_2 „ „ „ after „ „ after „ „ „ ,

B greatest breadth at midship section,

T ordinary draught „ „ „ ,

T_1 „ „ „ „ „ ex keel,

F girth of immersed „ „ ,

G mean girth to water line,

$\mathfrak{M}$ area of immersed midship section,

A „ „ rectangle circumscribed about immersed midship-section,

A_1 „ „ projection of emerged portion of ship upon plane of midship section,

O „ „ load water line,

S immersed surface of ship,

S_1 augmented „ „ „ ,

D displacement „ „ ,

W total resistance „ „ ,

W_f form „ „ „ ,

*) J. T. D'AUBUISSON De VOISINS. Traité d'hydraulique à l'usage des ingénieurs. Paris 1849.

**) J. WEISBACH. Lehrbuch der Ingenieur- und Maschinenmechanik. Bearbeitet von B. HERMANN. V. Aufl. II. Theil I. Abtheil. p. 230. Braunschweig 1882.

W_r friction resistance of ship,
W_w wave " " ",
W_v eddy " " ",
W_l air " " ",
W_p propeller " " ",
R (with various letters suffixed) the losses of work,
v speed of ship,
γ weight of unit volume of water,
g acceleration of gravity,
φ half angle of entrance,
φ_1 " " " run,
α_1 coefficient of fineness of fore part of load water line,
α_2 " " " " after " " " " ",
α_0 " " " " entire " " ",
β " " " " immersed midship section,
β_1 " " " " an auxiliary midship section,
c c_1 c_2 c_0, n n_1 n_2 n_0, f, i i_2, k k_1 k_2 k_3 k_4 k_c k_t, K K_1 K_2 K_0, m, t, u, u^1 are coefficients and constants whose signification is explained as they occur.

§ 38.

Calculation of the Horse-power by the Displacement Theory.

Classification of formulæ.

1) The various formulæ for calculating a ship's resistance by the displacement theory may be grouped as follows.

I. Formulæ which regard only the form-resistance,

a) CAMPAIGNAC'S,
b) MANSEL'S,
c) NYSTROM'S older formula;

II. Formulæ which regard the resistances of form and friction,

d) TREDGOLD'S,
e) GUÈDE and JAY'S,
f) THORNYCROFT'S,
g) NYSTROM'S later formula,
h) MIDDENDORF'S,
i) RIEHN'S,
k) ISHERWOOD'S;

III. Formulæ which regard other resistances, besides those of form and friction,

l) BOURGOIS'S,
m) DUPUY DE LÔME'S.

2) **Formulæ which regard only the form-resistance.** a. **Campaignac's formula***) is, as he says himself in the beginning of his work, only a new form of BORDA'S and MARESTIER'S and assumes the pressure exerted by the propeller against the water to vary as the square of the difference between the speed of the propeller and that of the ship and not, as in the old theory of BERNOUILLI, as the difference of the squares of these speeds. CAMPAIGNAC introduced into MARESTIER'S formulæ certain empirical coefficients which he determined from trials of good English and French steamers of that day, mostly having low-pressure engines by BOULTON and WATT.

Campaignac's formula.

His formula is

$$IHP = K\,\mathfrak{M}\,v^3 \quad \ldots\ldots\ldots\ldots\ldots \quad (173)$$

$\mathfrak{M}$ being given in sq. m. and v in knots, the coefficient K, which embraces all losses of effect from those in the cylinder to those of the propeller, has the following values,

Magnitude of the coefficients.

$K = 0.013$	for small engines up to			20 *NHP*	
$K = 0.012$	„ engines of	20	„	50	„
$K = 0.011$	„ „ „	50	„	160	„
$K = 0.010$	„ „ „	160	„	200	„
$K = 0.009$	„ „ „	200	„	300	„
$K = 0.008$	„ „ „	300	„	400	„
$K = 0.007$	„ „ „	400	„	500	„

This coefficient K is the reciprocal of the midship section coefficient C referred to in § 37, 9. The whole of CAMPAIGNAC'S formulæ are unsuitable for our modern marine engines.

Value of the formula.

4) The ship's resistance according to NAVIER, PONCELET, &c. is by Eq. 173

$$W = k\,\frac{\gamma}{2g}\,\mathfrak{M}\,v^2 = k_1\,\mathfrak{M}\,v^2 \text{ kg.} \quad \ldots\ldots\ldots \quad (174)$$

But here v is metres per sec. and k has the following values, according to the size and fineness of the ship,

$k =$ from 0.07 to 0.12 for sea-going steamers
$k =$ „ 0.14 „ 0.20 „ river „
$k =$ „ 0.25 „ 0.50 „ canal „

In spite of the indefiniteness of these coefficients, RIEHN states that the formula is often employed for river steamers. CAMPAIGNAC'S formula Eq. 173 is still much used in practice for approximate comparisons, but it must always be borne in mind that the application of it is limited to ships of the same type. For the coefficient K we must substitute the reciprocal of a midship-section coefficient C as obtained from the trial-trips of modern

*) A. CAMPAIGNAC. De l'état de la navigation par la vapeur. p. 10. Paris 1842.

steamers, the units being knots per hour for the ship's speed and *square metres for the midship-section.* These values for K may be got by taking the figures in col. 14 of the table on pp. 306 and 307, multiplying them by 0.093 and taking the reciprocal of the product.

Mansel's formula.

5) **Mansel's formula***) originates from the opinion that the midship-section formula

$$C = \frac{\mathfrak{M}\, v^3}{IHP}$$

or, which is the same thing, CAMPAIGNAC'S old formula

$$IHP = \frac{\mathfrak{M}}{C} v^3$$

in which C is known by comparison with similar existing ships, does not give accurate results, and that even for the same ship at different speeds it is very untrustworthy. He attempts to explain this by assuming that the expression

$$\frac{\mathfrak{M}}{C} v^3$$

does not give the indicated horse-power IHP, but only the net work due to the ship's resistance, — the effective horse-power $\mathfrak{E}HP$. In his opinion, this value must be augmented by the friction-work R_r of the engines as well as the work R_s lost through slip. He therefore writes

$$IHP = \frac{\mathfrak{M}}{C} v^3 + R_r + R_s$$

$$\mathfrak{E}HP = \frac{\mathfrak{M}}{C} v^3.$$

Useful performance of the engine.

6) MANSEL finds the effective horse-power as follows; if P and p are mean pressures on the HP and LP pistons respectively of a compound engine, and if the area of the LP piston is r times that of the HP, then the mean pressure reduced to the HP piston is $P + rp$. For multiple expansion engines the process is similar. Subtracting from this pressure the pressure f due to the friction-losses of the machinery, we get the useful mean pressure

$P + rp - f$ lbs per sq. inch engl.

whence of course the useful performance of the engine follows

$$\frac{\frac{\pi}{4} d^2\, 2s}{33\,000} N(P + rp - f) = \frac{d^2\, s}{21\,010} N(P + rp - f).$$

Ratio between useful pressure and speed of ship.

7) The useful performance of the engine is to the net work due to the ship's resistance as the rectilineal advance of the pro-

*) R. MANSEL. Proposition on the direct motion of steam vessels. Paper read before the mechanical section of the British Assn. Glasgow 1876.

peller is to the distance travelled by the ship. At v knots per hour the ship's distance per minute is 101,3 v ft. engl. and the advance of the propeller $N H c$, for N revs per min. and $H c$ its advance per rev. The coefficient c is found from $\frac{n_0 H}{101.3}$, where $n_0 = \frac{N}{v}$, so that we can now write

$$\frac{d^2 s}{21\,010} N(P + r p - f) \div \mathit{IHP} = N H c \div 101.3\, v$$

$$\mathit{IHP} = \frac{d^2 s}{21\,010} \frac{101.3\, v}{H c} (P + r p - f),$$

and substituting the above value of c

$$\mathit{IHP} = \frac{d^2 s}{21\,010} \frac{n_0}{c^2} (P + r p - f)\, v.$$

From this expression MANSEL derives the formula

$$\log (P + r p - f) = \log v + (\beta + \gamma)\, v + \log \frac{n_0}{l_0} \quad \ldots . \quad (175)$$

which exhibits a relation between useful pressure and speed of ship. In this formula, besides the known magnitudes,
β is a very small coefficient of the ship's speed, which may be positive or negative according to the nature of the propeller and is intended as a correction to the logarithms of the revolutions per mile,
γ, a coefficient the value of which ranges from 0.045 to 0.09 and varies with the ship's speed,
and $\frac{n_0}{l_0}$, a constant for the same ship under the same circumstances. For comparing dissimilar ships the expression becomes extremely complicated and is dependent upon the models, the propellers, and other conditions.

8) MANSEL*) afterwards devised another formula

Determination of the IHP.

$$\mathit{IHP} = b\, v \log^{-1} a\, v$$

or

$$\log \mathit{IHP} = \log b + \log v + a\, v \quad \ldots\ldots\ldots . \quad (176)$$

by the help of which, when the indicated horse-powers for two speeds not too closely approximated are known, he claims to be able to calculate the *IHP* at any other speed with great accuracy. To put this formula in a practical shape and shorten the working out he plotted a scale from which the approximate values of *log b* and *a* can be easily pricked off.

9) Both MANSEL'S formulæ are based on the circumstance of which he has given various examples, that a curve whose abscissæ are a ship's speeds, and the ordinates the logs of the corresponding

Value of the formulæ.

*) The speed of steam vessels. Engineering 1879. II. p. 429.

IHP's, must be *a straight line.* But in 1884 at the Institute of Engineers and Shipbuilders in Scotland Denny*) shewed that out of the trials made by the elder and younger FROUDE as well as out of thirty of his own, this straight line of MANSEL'S only occured in quite a few cases. At this rate MANSEL'S method is based on an erroneous hypothesis and any confidence in calculations made by it must be severely shaken.

Nystrom's older formula.

10) γ. *Nystrom's formula***) appears thus in the notaton adopted here,

$$W = 4\, \mathfrak{M} \sqrt{\frac{B^2}{B^2 + m\, L^2}}\, v^2\, \mathfrak{A} \text{ engl.} \quad \ldots\ldots\ldots (177)$$

Expressing all the dimensions in ft. engl. and v in knots per hour, we get the coefficient m from the argument

$$x = \frac{D}{\mathfrak{M}\, L}$$

(where D is the displacement in cubic ft. engl.) for which NYSTROM constructed the following table.

x	m	x	m	x	m	x	m
1	2	1	2	1	2	1	2
1.00	0.000	0.77	1.05	0.67	1.77	0.57	1.72
0.95	0.024	0.76	1.12	0.66	1.84	0.56	1.67
0.90	0.228	0.75	1.20	0.65	1.90	0.55	1.61
0.88	0.326	0.74	1.28	0.64	1.96	0.54	1.55
0.86	0.432	0.73	1.35	0.63	2.00	0.53	1.50
0.84	0.558	0.72	1.43	0.62	1.97	0.52	1.44
0.82	0.692	0.71	1.51	0.61	1.93	0.51	1.38
0.80	0.836	0.70	1.59	0.60	1.88	0.50	1.32
0.79	0.902	0.69	1.64	0.59	1.82	0.49	1.26
0.78	0.978	0.68	1.71	0.58	1.77	—	—

Determination of the IHP.

11) The *IHP* of the engine is obtained from Eq. 177 by multiplying W by the ship's speed in ft. engl. per second $= 1.689\, v$ which gives the work due to the resistance in foot-pounds engl. and dividing this product by 550. Inserting η, the efficiency of the engine, we get

$$IHP = \frac{1.689\, W v}{550\, \eta} \quad \ldots\ldots\ldots\ldots\ldots (178)$$

According to NYSTROM η varies between 0.66 for small to 0.72 for large engines.

*) Speed trials. Engineering 1885. I. p. 19.

**) J. W. NYSTROM. Pocket book of mechanics and engineering. Philadelphia 1869.

12) This older formula of NYSTROM'S is only valid for sea-going ships of considerable draught. Further, its results accord with experience only where there is a certain ratio of L to B. For short ships, i. e. for low values of $\frac{L}{B}$ the power comes out too high, for long ships too low. MIDDENDORF'S formula is derived from this but is an improvement upon it, as RIEHN'S is upon MIDDENDORF'S. **Value of the formula.**

II. Formulæ which regard the resistance both of form and friction.

13) d. **Tredgold's formula** *) was the first in which the resistances both of form and friction were taken into account. It is worked out for a prismatic model, sharpened at the ends and is expressed thus **Tredgold's formula.**

$$W = \frac{\mathfrak{B}\, v^2}{2}\,(2\,sin^3\,\varphi + sin^2\,\varphi_1) + f\,L\,F\,v^2\;\mathfrak{G}\text{s. engl.} \quad \ldots \;(179)$$

The first term represents the resistance of form, the second that of friction. With the dimensions in feet engl. and the speed of ship v in ft. per sec. TREDGOLD'S coefficient of friction $f = 0.0032$. The performance of the engine is determined as in Eq. 178.

$$IHP = \frac{W\,v}{550\,\eta} \quad \ldots\ldots\ldots\ldots\ldots \;(180)$$

the value of η being between 0.5 and 0.66, based upon the experience of that day with low-pressure paddle engines and efficient wheels.

14) According to RIEHN, TREDGOLD'S formula suffers from the want of a trustworthy foundation of experiment. It involves also the angles of entrance and run, elements of doubtful import. These angles, which have no signification save as the mean values of a series of angles taken at different depths, cannot be fixed on before the completion of the design. Even afterwards, sufficiently accurate measurement of them is impossible, especially with somewhat full waterlines and at the best it is questionable wheter these angles are of any material influence. **Value of the formula.**

15) e. **Guède and Jay's formula** **) was etablished from their trials of the French dispatch-vessel "Élorn" in 1861 (see § 37, 4 p. 303) and has the form **Guède and Jay's formula.**

$$W = k\,\mathfrak{B}\,v^2 + k_1\,\mathfrak{B}\,v^4 \text{ kg.} \quad \ldots\ldots\ldots\ldots \;(181)$$

The first term relates to form, the second to friction. For $\mathfrak{B}$ in sq. m and v in m per sec. the coefficients are

$$k = 2.6 \text{ and } k_1 = 0.15.$$

*) TH. TREDGOLD. The steam engine. London 1838. p. 293.

**) BERTIN. Notice sur la marine à vapeur de guerre et de commerce. Paris 1875.

BERTIN remarks these would become

$$k = 2{,}4 \text{ and } k_1 = 0{,}13.$$

if applied to the resistance found by FROUDE for the "Greyhound" at 8, 10, and 12 knots (see § 39, 15).

Determination of the IHP.

16) From Eq. 181 we may get

$$IHP = \frac{W v}{75\, u\, u^1} \quad \ldots \ldots \ldots \ldots \ldots \quad (182)$$

u and u^1 being the efficiencies of the engine and propeller respectively. LEDIEU says the average values of both are in general equal, i. e. $u = u^1$. The mean of the trial-trip results of forty ships of the French Navy mostly built in the seventies gives $u = u^1 = 0.8$. The "Èlorn" was tried with a four-bladed screw and at 8 to 12 knots shewed $u\,u^1 = 0.58$. On an average $u\,u^1$ is between 0.6 and 0.7 increasing with the efficiency of the machinery and with the speed v. The lower value is therefore to be taken for slow ships with low-pressure engines and the higher one for faster ships with compound engines. For the present high speeds and triples it would probably be higher still.

Value of the formula.

17) According to BERTIN, GUÈDE and JAY'S formula has been much used in France down to recent times on account of its great simplicity although the coefficients k and k_1 were determined from the trials of one ship only. For low speeds it is said to give very serviceable figures but not for high ones.

Thornycroft's formula.

18) f. **Thornycroft's formula** *) is derived from trial-trip results of various British war and merchant ships and has the following form

$$IHP = k\,v\left[f S \frac{3\,n}{2\,n + L} v^{1.7} + c\,\frac{n^{1/_8} + 3}{L^{1/_8} + 3} v^{3.7} \int sin^{2.5}\varphi\, d s\right] \quad . . \quad (183)$$

The first term represents the resistance of form, the second that of friction. For S and L in ft. engl. and v in knots the coefficients take the following values

$$k = 0.00451;\; f = 0.01264;\; c = 0.01587;\; n = 380.0,$$

ds is an element of the wetted surface,

φ the angle which this particular element makes with the middle line plane of the ship.

The *IHP* as given by this formula is based upon RANKINE'S assumption that the combined efficiency of the engine and propeller = 0.68.

Origin of the formula.

19) THORNYCROFT states that he was led to establish the above formula by the experience that for low speeds the ship's

*) J. J. THORNYCROFT. On the resistance opposed by water to the motion of vessels etc. Transactions of the Institution of Naval Architects. London 1869. p. 144.

resistance does not vary with the square of the speed. He divided the total resistance into two parts and by ELDER'S advice based the resistance of form upon experiments on the motion of water in pipes, from which he inferred that the resistance increased at a slower rate than as v^2, his trials giving $v^{1.7}$. As however the total resistance in the trial-trip results worked out by him, which were obtained at various speeds, increases faster than as v^2, he came to the conclusion that the residuary resistance must rise in a very high ratio of v, which he takes at $v^{3.7}$.

20) Even if we admit the correctness of the preceding suppositions in many cases where the formula gives useful results, we cannot help agreeing with SCOTT RUSSEL'S remark in the discussion of the formula at the Institution, that such a complicated expression involving difficult and tedious calculation of the quantity under the integral sign does not correspond with practical requirements. Value of the Formula.

21) g. **Nystrom's later formula***) with the notation adopted here is stated thus, Nystrom's later formula.

$$W = 2.858\, v^2 \left[\mathfrak{M}\, (0.9 \sqrt{\sin^3 2\,\varphi} + 0.1 \sqrt{\sin^3 2\,\varphi_1}) \right]$$

$$+ \frac{k}{\sqrt[4]{L}} (O - 2\,(\mathfrak{M} - L\, T_1)) \sqrt{\frac{D}{B\, L\, T_1}} \text{ ℔s. engl.} \quad \ldots \quad (184)$$

The first portion relates to form, the second to friction and all measurements are in English units, v being in knots. The mean angle of entrance $2\,\varphi$ and the mean angle of run $2\,\varphi_1$ are determined thus

$$tg\, 2\,\varphi = \frac{\mathfrak{M}\, t}{L_1\, T_1};\quad tg\, 2\,\varphi_1 \frac{\mathfrak{M}\, t}{L_2\, T_2}$$

t is a coefficient dependent upon the ratio of the cylinder, whose cross section is the midship section, to the displacement, for which NYSTROM has worked out a series of tables,

k is the coefficient of friction of the wetted surface of the ship in ℔s. per sq. ft. engl.; its value is

for polished metallic surfaces	$k = 0.003$
„ ordinary copper sheathing	„ $= 0.005$
„ smooth wood surfaces	„ $= 0.007$
„ rough cast iron or bronze surfaces	„ $= 0.009$
„ foul bottoms	„ $= 0.015$
„ „ „ with grass and shells	„ $= 0.020.$

22) NYSTROM determines the *IHP* by putting the bracketed expression in the first summand in his formula (184) + the whole of the second summand equal to X and writing Determination of the *IHP*.

$$W = 2.858\, v^2\, X \text{ ℔s engl.}$$

*) Journal of the Franklin Institute. Philadelphia. February 1882. p. 139.

Multiplying W by the ship's speed in ft. engl. per sec., 1.689 v, we get the work of resistance in footpounds = 1.686 Wv and by Eq. 178

$$\mathit{EHP} = \frac{2.858\, v^2\, X \times 1.689\, v}{550} = \frac{v^3\, X}{113.82}.$$

The IHP is then supposed to be about 50% greater than the effective, thus

$$IHP = 1.5\ \mathit{EHP} = \frac{v^3\, X}{75.88} \quad \ldots\ldots\ldots\ldots (185^a)$$

NYSTROM gives the formula for IHP in his pocket book*) as

$$IHP = \frac{Wv}{224.5} \quad \ldots\ldots\ldots\ldots (185^b)$$

which expression may be derived from Eq. 178 if η is taken as = 0.68.

Value of the Formula.

23) NYSTROM'S later formula is adapted to his parabolic system of ship design and therefore only accurate for ships built according to it. The formula is much more complicated than THORNYCROFT'S. The determination of the coefficient t which relates to the mean angles of entrance and run is particularly tedious unless the ship is designed on the parabolic system. As besides, the effect of these angles upon the calculation is always highly vague and doubtful (see 14) it is hardly to be expected that this later formula will give any more accurate results than the earlier one, the more so as it assumes the useful effect of large and small engines to be equal. It is not likely to be much used in practice on account of the voluminous calculations required.

Middendorf's Formula.

24) g. **Middendorf's Formula****) is derived from NYSTROM'S earlier one (§ 38, 10 p. 318) which only regards the form-resistance, by altering the coefficients and inserting the friction-resistance, as based on trial-trip results, mostly obtained in the seventies from steamers built at the "Weser" yard in Bremen. In determining the friction-resistance the ship's skin was not assumed to be perfectly clean and smooth but only in such moderately good condition as would occur in reality. The formula is

$$W = 11 \frac{\mathfrak{B}\, B}{\sqrt{B^2 - m\, L^2}}\, v^{2.5} - k\, S\, v^2 \text{ kg.} \quad \ldots\ldots (186)$$

The first term relates to form, the second to friction. Expressing v in metres per sec. and all the other magnitudes on

*) J. W. NYSTROM. Pocket book of mechanics and engineering. Philadelphia 1882. p. 449.

**) M. RÜHLMANN. Hydromechanik. Hannover 1880. p. 752.

the metrical system, the coefficient $k = 0.17$. With very smooth skin such as copper the friction resistance becomes $0.17\,S\,v^{1.85}$. For most ships of ordinary form $m = 2$, otherwise its value must be taken from the following table by the help of the argument $x = \frac{D}{\mathfrak{B}\,L}$.

x	m	x	m
0.7 and less	2.00	0.80	1.62
0.71	1.99	0.81	1.50
0.72	1.98	0.82	1.42
0.73	1.96	0.83	1.32
0.74	1.93	0.84	1.18
0.75	1.89	0.85	1.06
0.76	1.85	0.86	0.90
0.77	1.81	0.87	0.74
0.78	1.75	0.88	0.55
0.79	1.69	0.89	0.31
		0.90	0.02

25) MIDDENDORF determines the IHP from the resistance W as follows. As the total resistance W is equal to the pressure exerted by the water against the propeller, we must have **Determination of the IHP.**

$$W = \varepsilon\, a\, (u - v)^2,$$

where a is the disc-area of the screw, or the combined surface of two paddle-floats (or the area of the delivery outlets of hydraulic propellers), and u is the speed of the propeller. Then

$$u = v \times \sqrt{\frac{W}{\varepsilon\, a}}$$

and the performance of the engine

$$\mathfrak{HP} = \frac{W u}{75} = \frac{W}{75}\left(v \times \sqrt{\frac{W}{\varepsilon\, a}}\right).$$

If the coefficient ε is taken at 160 and the efficiency of the engine $= \eta$

$$IHP = \frac{W}{75\,\eta}\left(v \times \sqrt{\frac{W}{160\,a}}\right) \quad \ldots\ldots\ldots\ldots \quad (187)$$

The value of η is to be taken from the table on p. 324 according to the IHP of the engine.

26) RIEHN states that MIDDENDORF'S formula gives very good results for average proportions of length to breadth especially for short vessels with very fine lines. **Value of the formula.**

It is also trustworthy for long and full ships, but for very long and sharp ships the values are said to come out rather too high.

IHP	η	IHP	η
30 — 40	0.61	800 — 900	0.76
40 — 60	0.62	900 — 1000	0.77
60 — 80	0.63	1000 — 1200	0.78
80 — 100	0.64	1200 — 1400	0.79
100 — 150	0.65	1400 — 1600	0.80
150 — 200	0.66	1600 — 2000	0.81
200 — 250	0.67	2000 — 2500	0.82
250 — 300	0.68	2500 — 3000	0.83
300 — 350	0.69	3000 — 3500	0.84
350 — 400	0.70	3500 — 4000	0.85
400 — 450	0.71	4000 — 5000	0.86
450 — 500	0.72	5000 — 6000	0.87
500 — 600	0.73	6000 — 7000	0.88
600 — 700	0.74	7000 — 8000	0.89
700 — 800	0.75	8000 and above	0.90

It is certain that it greatly excels all other approximate formulæ in exactness and is therefore to be highly commended for practical use.

Riehn's formula

27) h. **Riehn's formula** *) is based partly on elaborate theoretical investigations and partly on practical data. The first term of it refers to the resistance of form including eddy and wave-resistance, the second term to the friction-resistance. The principal dimensions and coefficients of fineness of the ship are assumed to be known, as they must be for a proposed steamer, and RIEHN very wisely gives different formulæ for sea-going ships which travel in limitless water and for river and canal boats where the channel is more or less confined.

Resistance of ordinary sea-going ships.

28) *In ordinary sea-going ships with fine or moderately full water-lines* for which α_1 or α_2 lie between 0.55 and 0.79 and β between 0.5 and 0.9 and

$$\beta_1 = \beta\left(1.1 - 0.125\,\beta\,\frac{B}{T_1}\right),$$

$$W = 20\,A\left[\left(\frac{B}{2\,L_1}\right)^2\left(C_1 - 1\right)K_1 + \left(\frac{B}{2\,L_2}\right)^2\left(C_2 - 1\right)K_2\right]v^{2.5} - 0.127\,\frac{L}{B}\,A\left(2 + \frac{\alpha_0 B}{T_1}\right)v^{1.83}\ \text{kg.} \quad (188^a)$$

In this expression

$$C_1 = \frac{n_1^3}{3\,n_1 - 2} \times \frac{1.1}{1 - n_1^2\left(\frac{B}{2\,L_1}\right)^2}$$

*) W. RIEHN. Die Berechnung des Schiffswiderstandes. Hannover 1882.

for the fore part of the load water line and

$$C_2 = \frac{n_2^3}{3\,n_2 - 2} \times \frac{1.1}{1 - n_2^2 \left(\frac{B}{2\,L_2}\right)^2}$$

for the after part of the load water line; further

$$K_1 = a + \frac{k}{C_1 - 1}$$

$$K_2 = a + \frac{k}{C_2 - 1}$$

in which again

$$k = 1 - \frac{3}{m+1} + \frac{3}{2\,m+1} - \frac{1}{3\,m+1} \text{ and}$$

$$a = \frac{1}{3} - \frac{19}{3} \times \frac{1}{m+1} + \frac{3}{2\,m+1} - \frac{1}{3\,m+1} + \frac{6}{m+2} + \frac{2}{3\,m+2} - \frac{3}{m+3} + \frac{3}{2\,m+3}.$$

The values of m and n are found from

$$m = \frac{\beta_1}{1 - \beta_1}$$

$$n = \frac{\alpha}{1-\alpha};\ n_1 = \frac{\alpha_1}{1-\alpha_1};\ n_2 = \frac{\alpha_2}{1-\alpha_2},$$

and thus all the magnitudes for the above formula are determined, the dimensions being all metric and v expressed in m per sec. as in all the following formulæ of RIEHN'S.

If $n_1 = n_2 = n_0$, and therefore $C_1 = C_2 = C_0$ and $L_1 = L_2$, we get

$$W = 40\,A\left(\frac{B}{L}\right)^2\left(C_0 - 1\right)K_0\,v^{2.5} + 0.127\,\frac{L}{B}\,A\left(2\frac{\alpha_0\,B}{T_1}\right)v^{1.83}\text{ kg.} \quad (188^b);$$

in this expression

$$C_0 = \frac{n_0^3}{3\,n_0 - 2} \times \frac{1.1}{1 + n_0^2\left(\frac{B}{L}\right)^2}$$

$$K_0 = a + \frac{k}{C_0 - 1}.$$

This formula 188[b] can be used for an approximation in case the proportions of α_1 and α_2 to the whole coefficient of fineness α_0 are not known.

29) *For ordinary sea-going ships with very full load-water-line in the run*, where α_1, β, and β_1 are determined as in 28) but where α_2 lies between 0.8 and 0.87 **Resistance with full run.**

$$W = 20\,A\left[\left(\frac{B}{2\,L_1}\right)^2\left(C_1 - 1\right)K_1 + \left(\frac{B}{2\,L_2}\right)^2\left(\alpha_2 C_2 - 1\right)K_2\right]v^{2.5}$$

$$+ 0.127\,\frac{L}{B}\,A\left(2 + \frac{\alpha_0\,B}{T_1}\right)v^{1.83}\text{ kg.} \ldots\ldots\ldots \quad (189)$$

C_1, C_2, K_1, a and k have the same signification as in 28), but

$$K_2 = a + \frac{k}{a_2 C_2 - 1}.$$

For a_2 we can put

$$a_2 = 0.9 \text{ if } \alpha_2 = 0.80 \text{ to } 0.81,$$
$$a_2 = 0.8 \ \text{„} \ \alpha_2 = 0.82 \ \text{„} \ 0.83,$$
$$a_2 = 0.7 \ \text{„} \ \alpha_2 = 0.84 \ \text{„} \ 0.87.$$

Resistance of ordinary river steamers.

30) *River steamers with flat bottom, but with rounded bilge in the entrance and run.* Besides the dimensions, α_1 and α_2 are given, but β, β_1, and m do not require to be regarded.

$$W = 20\,A\left[i_1\,C_1\left(\frac{B}{2\,L_1}\right)^2 + i_2\,C_2\left(\frac{B}{2\,L_2}\right)^2\right]v^{2.5} + 0.153\,\frac{L}{2\,B}\,A\left(2 + \frac{\alpha_0\,B}{T_1}\right)v^{1.83}\,\text{kg.} \quad (190^a)$$

Here we must substitute C_1 and C_2 according to Eq. 188^a and put

$$i_1 = \frac{1}{2} + \frac{1}{2\,C_1};\; i_2 = \frac{1}{3} + \frac{2}{3\,C_2}.$$

If $\alpha_1 = \alpha_2 = \alpha_0$ and therefore $C_1 = C_2 = C_0$ and $L_1 = L_2$, then

$$W = 20\,A\,C_0\left[\left(\frac{B}{L}\right)^2\left(i_1 + i_2\right]v^{2.5} + 0.153\,\frac{L}{B}\,A\left(2 + \frac{\alpha_0\,B}{T_1}\right)v^{1.83}\ \text{kg.} \quad (190^b)$$

Resistance of shallow-draught river steamers.

31) *For river and canal steamers of very shallow draught* and of almost angular form at the bilge throughout their length, when they are steaming in a sufficiently deep and wide channel

$$W = 20\,A\left[C_1\left(\frac{B}{2\,L_1}\right)^2 + C_2\left(\frac{B}{2\,L_2}\right)^2\right]v^{2.5} + 0.17\,\frac{L}{B}\,A\left(2 + \frac{\alpha_0\,B}{T_1}\right)v^{1.83}\,\text{kg.} \,(191^a)$$

C_1 and C_2 have the same signification as in Eq. 188^a.
If $C_1 = C_2 = C_0$ and $L_1 = L_2$ then

$$W = 40\,A\,C_0\left(\frac{B}{L}\right)^2 v^{2.5} + 0.17\,\frac{L}{B}\,A\left(2 + \frac{\alpha_0\,B}{T_1}\right)v^{1.83}\ \text{kg.}\ . \quad (191^b)$$

Ships with parallel sides.

32) In all vessels of the above types, having parallel middle-bodies, L_1, α_1, and n_1 refer only to the sharpened part of the fore-body, and L_2, α_2, and n_2 to the sharpened part of the after-body. In the formulæ in which the resistance of form is directly determined by α_0, n_0, and C_0, $\frac{B}{L}$ is not to be used in the first term (on the right), but $\frac{B}{2\,L_1}$ or $\frac{B}{2\,L_2}$ instead; in the second term (on the right) $\frac{L}{B}$ must however always be used.

Air-resistance.

33) RIEHN has also determined the air-resistance and puts it at

$$W_l = 0.16\,A_1\,v_1^2\ \text{kg}$$

where A_1 is the projection of the unimmersed portion of the ship upon the plane of the midship section

and v_1 the relative velocity in m per sec. at which the wind strikes this plane. In a calm $v_1 = v$.

He however omits this expression in the formulæ for the total resistance for the reasons given in § 37, 19.

Determination of the *IHP*.

34) According to RIEHN the *IHP* is

$$IHP = \frac{Wv}{75\,\eta} \quad \ldots \ldots \ldots \ldots \quad (192)$$

in which η is to taken from this table.

For $\frac{Wv}{75}$ =		12	to	15 *HP*	η =	0.26	to	0.30.
,,	,,	15	,,	20 ,,	,,	0.20	,,	0.33.
,,	,,	20	,,	30 ,,	,,	0.33	,,	0.35.
,,	,,	30	,,	60 ,,	,,	0.35	,,	0.40.
,,	,,	60	,,	90 ,,	,,	0.40	,,	0.45.
,,	,,	90	,,	120 ,,	,,	0.45	,,	0.50.
,,	,,	120	,,	400 ,,	,,	0.50	,,	0.55.
,,	,,	400	,,	1000 ,,	,,	0.55	,,	0.60.
,,	,,	1000	,,	2000 ,,	,,	0.60	,,	0.65.
,,	,,	2000	,,	4000 ,,	,,	0.65	,,	0.70.

For river steamers in shallow water, the above values are to be reduced by from 0.08 to 0.1, owing to the loss of efficiency from the incomplete immersion of the propeller.

Value of the formula.

35) RIEHN'S formula (in the examples which he quotes) gives results which correspond with the trial-trip performances, and is therefore one of the most accurate. Although it requires voluminous calculations, even with the tables he has worked out for drawing-office use, it has of late years become more and more popular in Germany especially for river-steamers.

Isherwood's Formula.

36) i. **Isherwood's formula** *) only gives the friction-resistance. By subtracting this from the total resistance found as explained below, the form-resistance of the ship is obtained. ISHERWOOD established this formula on the results of his trials with steam launch No. 4 of the U. S. Navy, carried out at Mare Island California in 1869 and 70. He assumed that the condition of the skin remained about the same in all the trials and that the friction-resistance varied as the speed squared. He says further that an exact calculation of the friction-resistance is impossible on account of the ever varying curvature of the ship's immersed surface, the relative velocity of the surrounding water with regard to this being nowhere so great as the ship's speed except at such parts of her as are parallel to the longitudinal axis. This resistance can however be approximately ascertained upon the

*) Journal of the Franklin Institute. Philadelphia 1875.

assumption that the velocity of this immersed surface is less than the velocity of the ship in the ratio of the base to the hypothenuse of a right-angled triangle the base of which is the half length of the ship on the waterline and its perpendicular the half breadth on the waterline. According to his previons trials the friction-resistance is 0.45 lbs. per sq. ft. of wetted surface at a speed of 10 ft. per sec. and varies as the speed squared. Calling v_1 the speed of the wetted surface in ft. engl. per sec. determined by the above method and S its area in sq. ft. engl., the friction-resistance

$$W_r = \frac{0.45\, S\, v_1^{\,2}}{10^2} = 0.0045\, S\, v_1^{\,2} \text{ lbs. engl. } \ldots\ldots (193)$$

Determination of the total resistance.

37) In his trials with the launch No. 4 ISHERWOOD ascertained the total resistance W by inserting a dynamometer in the screw-shaft, whereas in his subsequent experiments (see below in the § on Experience with Screws) with the steam yacht "Lookout" and the steamer "Dispatch", the total resistance was calculated by the following method derived from the trial-results of the launch which in her case gave the total resistance as only 0.275 % greater than that recorded by the dynamometer.

Method of conducting the trials

38) ISHERWOOD determined, a) *The indicated horse-power IHP* as usual. b) *The work required to drive the engine when empty*, i. e. *disconnected from the shaft* IHP_1. The two cylinder single-expansion engine of the launch required 2 lbs. per sq. in. engl. mean pressure to drive it at the usual revolutions. The same pressure on *both* pistons was required for the two-cylinder compound engine of the yacht "Lookout"; in one case (with screw E) it was 2.7164 lbs. per sq. in. reduced to the LP piston. For the two-cylinder single-expansion engine of the steamer "Dispatch" it was $1^3/_4$ lbs. per sq. in.

c) *The work lost in the loaded engine due to friction caused by the load* IHP_r is put in all the trials as

$$IHP_r = 0.075\,(IHP - IHP_2).$$

It therefore amounts to $7^1/_2$ % of the difference between the gross IHP and that part of it required to drive the engine running disconnected. ISHERWOOD thus estimates the loss of work from friction due to the load (see § 30, 30 p. 259) much higher than THURSTON, but still only rather more than half as high as FROUDE, who takes it, like VOELCKERS at 13 % of the gross IHP_1 i. e. $IHP_r = 0.13\, IHP$.

d) *The work due to water-friction of the screw* IHP_v is found in the same way as the friction-resistance of the ship, see 36. The developed surface of both sides of the screw blades S_s is taken in sq. ft. engl. and its helical velocity v_2 in ft. per sec. for every

different speed of ship. As the frictional resistance is 0.45 ℔ per sq. ft. engl. at a velocity of 10 ft. per sec. and varies as the velocity squared, the work due to water-friction of the screw,

$$IHP_v = \frac{0.0045\, S_s}{550} v_2^2 = 0.000008181\, S_s v_2^2.$$

e) *The work lost by slip* IHP_s. In the following expression

$$IHP - (IHP_l + IHP_r + IHP_v),$$

the difference is the work $IHP_s + IHP_w$, if this sum represents the work corresponding to the ship's total resistance. These two works are to each other as the portion of the screw's advance lost by slip is to its total advance. For instance if the slip is 15 % of the speed of screw, i. e. 0.15 (Revolutions × Pitch), the work

$$IHP_s = 0.15\,[IHP - (IHP_l + IHP_r + IHP_v)].$$

f) *The work due to the ship's total resistance* IHP_w would in this instance be

$$IHP_w = 0.85\,[IHP - (IHP_l + IHP_r + IHP_v)].$$

If, as in the case of the steam launch, it is determined by means of a dynamometer and the recorded pressure is P at speed v ft. engl. per sec., we have

$$IHP_w = \frac{P v}{550}.$$

39) The table on p. 332 exhibits the various quantities of mechanical work and their ratios as determined by ISHERWOOD for six different vessels. The percentages refer to the nett performances of the engines, i. e. the gross IHP less the disconnected IHP. Only the last quantity in the table relates to the proportion between the gross IHP and the work due to the ship's total resistance. Of all the trials with eight different screws (see further on in the § on experience with screw propellers) to which steam launch No. 4 was subjected, only the trial with screw B is reported because the slip of this screw about corresponds with that of the other two vessels. For the "Lookout" the results of the runs with seven different propellers are given as they would have been if she had steamed 10 knots with screw E, her bottom being coppered, as ISHERWOOD considers the results obtained under these particular conditions were the most trustworthy of all. "Despatch" only made the one trial run as given. The vedette, built by HERRESHOFF was tried with three screws. With the first screw three progressive runs were made, the fastest of which is recorded here, no indicator cards were taken with the second screw, and the results with the third screw were inferior to those with the first. "Siesta" also built by HERRESHOFF was subjected to twelve trials, and their mean results as calculated by ISHERWOOD are given. Steam launch No. 8 built for the Commander of the New York Navy Yard **Table of resistances.**

made eight progressive runs, the results of the fastest of which are given in the table on p. 332.

Value of Isherwood's trials.

40) From that table the useful effect of the engines η appears quite unusually high compared with FROUDE and DENNY'S results, especially for such small engines (see § 39, 24), whereas the friction-work IHP_r is very low in comparison with THURSTON'S trials. In spite of these discrepancies from the universally recognized results of FROUDE and DENNY, these trials of ISHERWOOD'S are nevertheless of great interest, particularly on account of his close determination of the ship's resistance at low speeds. The trials of the steam launch shewed that the resistance up to a speed of 6 knots increased at a lower rate than as the speed squared, from 6 to 7 approximately at that rate, but that at 8.5 it already increased in a 20% greater ratio. ISHERWOOD also determined this ratio for the vedette*) and the steam-cutter**) and found as follows in the table

Boat	Speed in knots	Thrust of screw in lbs. english	Power of speed with which resistance varies
Vedette	6.6803	198.8900	
	9.1548	482.2400	2.8107
	10.9700	1286.5700	5.4250
	11.7422	1589.3000	3.1064
Steam-cutter	4.2705	144.5433	
	5.5843	296.4816	2.6783
	6.5336	405.5844	2.0255
	6.9142	472.1184	2.5780
	7.5790	599.4837	2.6016
	8.5247	924.5317	3.6842
	8.8525	1125.1129	5.2038

Lewicki's approximate formula.

41) LEWICKI***) constructed a curve based on ISHERWOOD'S experiments with the steam launch No. 4, in which the abscissæ were the speeds and the ordinates the corresponding resistances. This curve is of an S-shape, which arises from the little craft trimming more and more by the stern with increasing speed. As she was very sharp forward and had a very flat transom,

*) Report made to the bureau of steam-engineering, navy department, August 9. 1882. Washington 1882. P. 41.

**) Journal of the american society of naval engineers. February 1889. P. 12.

***) L. LEWICKI. Vorträge über Maschinenbau; Dampfschiffbau. Herausgegeben vom Maschinen-Techniker-Verein des Polytechnikums zu Dresden. 1882. Sheet 116 & 117.

the resistance at first of course increased faster than proportionally to the speed alone. ISHERWOOD explains this circumstance thoroughly in his dissertation upon the trial-results of the steam-cutter. He expresses the opinion that for every ship there is a speed at which the water can no longer sufficiently rapidly fill up the void left by the progress of the hull, so that her stern sinks until it finds the necessary support from the surrounding water, thus increasing the resistance so considerably. In fact he says the same thing in other words as was pointed out in § 37, 27, viz. that the economical speed is exceeded when the stern wave comes astern of the ship. LEWICKI draws through the S-shaped curve an easier curve, the ordinates of which up to speeds of 6.5 knots are higher, about the same at 6.5 to 7, and after that lower. The ordinates of this curve of resistance are calculated by the empirical formula

$$W = 0.06064\ \mathfrak{M}\ v^{2.09} + 0.0045\ S\ v^2\ \text{lbs. engl.} \quad \ldots\ldots (194)$$

The first term represents the form-resistance, the second the friction-resistance calculated from Eq. 193. but in so far approximately that the ship's speed v is substituted for ISHERWOOD'S v_1.

42) To get the $IH\!P$ by help of this formula we must multiply the resistance W by the ship's speed v in ft. engl. per sec. and divide by 550 η **Determination of the $IH\!P$.**

$$IH\!P = \frac{Wv}{550\,\eta},$$

where η has the values found by ISHERWOOD in his trials to vary between the somewhat wide limits of 0.5 to 0.75. It is to be remarked that this formula, as may be supposed from its origin, can only give correct figures for small vessels at low speeds and with unusually efficient engines.

43) **III. Formulæ which regard other resistances besides those of form and friction.** **Bourgois's formula.** 1. **Bourgois's formula** *) is based on the experiments made by D'ALEMBERT, BOSSUT, and CONDORCET of the French Académie des Sciences at Paris in 1775 and 1778 for the government with the object of determining the resistance of floating bodies and models of ships. The formula also makes use of the results of some similar experiments with small vessels made at London from 1793 to 1798 by BEAUFOY. It has the following form

$$W = k_1\ \mathfrak{M}\ v^2 + k_2\ B\ v^4 + k_3\ S\ v + k_4\ S\ v^2\ \text{kg.} \quad \ldots\ldots (195)$$

The first term is the resistance of form, the second the wave-resistance, the third a resistance called by BOURGOIS the *re-*

*) M. BOURGOIS. Mémoire sur la résistance de l'eau au mouvement des corps et particulièrement des bâtiments de mer. Paris. 1857.

Table of Isherwood's Results.

No.		Steam launch No. 4.		"Lookout".		"Dispatch".		Vedette.		"Siesta".		Steam-cutter.	
1	Date of trial	1869—1870		January-October 1880		14. January 1880		23. June 1881		22. June 1882		22. November 1888	
2	Displacement of ship in tons engl.	23.3053		42 8700		552.2500		7.4430		63 8300		17.3875	
3	Ship's Speed v in knots	8.5000		10.0000		10.7500		11.7422		11.0000		8 9015	
4	Slip of screw in °/₀ of its speed	16.1500		16.0000		15.0914		18.0000		27.4100		35 8000	
5	Indicated horsepower of engines IHP	39.3303		79 5393		402 7950		95 2425		140.3365		60.4848	
6	Disconnected power IHP_l	1.1420		9 2265		32.3700		11.4466		11.9437		3.0940	
7	Nett power of engines $IHP - IHP_l$	38.1883	= 100	70.3128	= 100	370.4249	= 100	83.7559	= 100	128.3927	= 100	57 3908	= 100
8	Power absorbed by engine friction IHP_r	2.8641	7.5000 °/₀	5.2734	7.5000 °/₀	27.7819	7 5000 °/₀	6.2817	7.5000 °/₀	9.6294	7 5000 °/₀	4 3043	7 5000 °/₀
9	Power absorbed by friction of screw IHP_v	1.5978	4.1900 °/₀	5.3474	7.6051 °/₀	25.6238	6.9147 °/₀	7.5209	8.9800 °/₀	11.0551	8 6100 °/₀	4.5508	7 9296 °/₀
10	Power absorbed by slip IHP_s	5.4468	14.2600 °/₀	9.5507	13.5832 °/₀	47 8426	12.9156 °/₀	12.5916	15 0300 °/₀	30 1887	23 5100 °/₀	17 3757	30 2762 °/₀
11	Power due to ship's resistance IHP_w	28 2796	74.0500 °/₀	50.1413	71.3117 °/₀	269 1766	72.6670 °/₀	57 3617	68 4900 °/₀	77 5194	60.3800 °/₀	31.1598	54 2942 °/₀
12	Ratio η of power due to ship's resistance IHP_w to indicated horse-power of engines IHP	0 74		0 63		0 67		0.60		0.55		0.51	

sistance of cohesion which the water particles oppose to their separation, the fourth term is the friction-resistance. If we put

$$W = k_t\, \mathfrak{D}\, v^2$$

we can also write

$$k_t\, \mathfrak{D}\, v^2 = k_1\, \mathfrak{D}\, v^2 + k_2\, B\, v^4 + k_3\, S\, v + k_4\, S\, v^2$$

or the coefficient of total resistance

$$k_t = k_1 + k_2 \frac{B\, v^2}{\mathfrak{D}} + \frac{k_3\, S}{\mathfrak{D}\, v} + k_4 \frac{S}{\mathfrak{D}}.$$

As BOURGOIS states that the form-resistance, and friction-resistance are independent of the speed, he combines them in one coefficient

$$k_c = k_1 + k_4 \frac{S}{\mathfrak{D}}$$

and then writes

$$k_t = k_c + k_2 \frac{B\, v^2}{\mathfrak{D}} + k_3 \frac{S}{\mathfrak{D}\, v}\ \text{kg.} \quad \ldots\ldots\ldots \quad (196)$$

If in this formula we express v in m per sec. and all the other magnitudes on the metrical system, the coefficients k_c, k_2, and k_3 assume the following values.

k_c = from 1.8 to 2.2, for steamers whose length is $5^1/_2$ times their beam and upwards.

k_2 = „ 0.12 „ 0.16, for steamers of fine lines, such as fast despatch boats, up to ships of fuller lines more resembling sailing ships.

k_3 = „ 0.08 „ 0.10, for steamers which are either coppered or otherwise rendered perfectly smooth and clean, up to those with iron skin several months in the water and covered with a slight growth.

44) BOURGOIS also determined the air-resistance, but does not in- Air-resistance.
corporate it in his formula for the general reasons given in § 37, 19. According to his experience with French war-ships, the coefficient of total resistance k_t is to be augmented by from 0.2 to 0.45 if the air-resistance in a calm is to be taken into account, the lower limit referring to ships of the line and the higher to small despatch-boats. The coefficient of total resistance k_t with the air-resistance included has the following general values for the respective speeds and types of ships.

$k_t = 3.0$	kg	for ships of the line	and	11	knots	speed,
$k_t = 3.4$	„	„ frigates	„	„	„	„ ,
$k_t = 3.8$	„	„ corvettes	„	10	„	„ ,
$k_t = 4.6$	„	„ despatch-boats	„	„	„	„ ,
$k_c = 5.5$	„	„ gun-boats	„	„	„	„ .

Later coefficients of total resistance.

45) For later ships and in particular for the trial-trips of 40 different vessels of the French Navy held between 1869 and 1879 ANTOINE*) gives the coefft. of total resistance k_t as

$$k_t = \frac{v}{\sqrt[3]{\mathfrak{M}}} + t \quad \dots \dots \dots \dots \dots (197)$$

The constant t has the values

$t = 2.5$ for iron ships of ordinary lines,
$t = 2.0$ „ „ „ „ fine „ ,
$t = 1.5$ „ „ „ without masts,
$t = 2.0$ „ coppered „ of ordinary lines,
$t = 1.5$ „ „ „ „ fine „ ,
$t = 1$ „ „ „ without masts.

Determination of the *IHP*.

46) The *IHP* is calculated from the resistance,

$$IHP = \frac{W v}{75\,\eta},$$

or

$$IHP = \frac{k_t\,\mathfrak{M}\,v^3}{75\,\eta} \quad \dots \dots \dots \dots \dots (198^a)$$

η being as before the efficiency of the engine. In applying this formula it is usual, at the French Admiralty, to transform the expression thus

$$\frac{\eta}{k_t} = \delta$$

whence follows

$$IHP = \frac{\mathfrak{M}\,v^3}{75\,\delta} \quad \dots \dots \dots \dots \dots (198^b)$$

According to LEDIEU'S tables in which are collated the trial-trip results of 35 old**) and 30 recent***) ships of all types, the value of δ varies on an average between

$\delta = 0.08$ for small ships of low power and
$\delta = 0.15$ „ larger „ „ higher „ .

Value of the Formula.

47) BOURGOIS'S formula (195) for the resistance, the coefficients of which were all determined for ships of low speed (up to 11 knots) is no longer closely applicable to modern fast steamers. Even formerly it never obtained the confidence of designers, being reputed to give rather too low values, nevertheless it indisputably takes the first place among the older formulæ. The formula 198^b can however be profitably applied to the computation of the *IHP* for a proposed engine, if, as is done at the French Admiralty, the coefficient δ (coefficient d'utilisation) is worked out for every new ship after her trials, and placed in a group corresponding to her particular type. — To

*) CH. ANTOINE. Calculs des propulseurs héliçoïdaux. Paris. 1880.

**) A. LEDIEU. Atlas du traité élémentaire des appareils à vapeur de navigation. Paris. 1865. Table J. and K.

***) A. LEDIEU. Atlas des nouvelles machines marines. Paris 1879. Table B.

guard against confusion between the coefficient δ and the coefficient of performance C (see § 37, 9) it is to be noted that the former is calculated with v in m. per sec. and the latter with v in knots per hour.

48) m. **Dupuy de Lôme's formula***) is the fruit of a long series of trials of the most divergent types of ships carried out by this celebrated constructor in the years from 1841 to 1856, sometimes at sea and sometimes in TOULON harbour, but always in calm weather. His empirical formula, which gives the resistances almost exactly as they were observed by him, has the form **Dupuy de Lôme's Formula.**

$$W = k\,\mathfrak{M}\,(v^2 + 0.145\,v^4) + k_1\,S\sqrt[3]{v}\ \text{kg.} \quad \ldots\ldots \quad (199)$$

The first term refers to form, the second to friction. If we express v in m per sec., $\mathfrak{M}$ and $S = GL$ in sq. m then

k is a coefficient dependent on the form of the ship which diminishes at the same rate as the square-root of the radius of curvature of the bow and buttock lines is increased. It also decreases with the mean angle of entrance, — about 15 % if the latter is reduced from 45^0 to 15^0, or about 0.5 % per degree between these limits. At the trials of the line of battle ship "Napoléon" k was 1.96 kg.

k_1 a coefficient of smoothness of bottom. For quite clean and faultlessly smooth copper it is to be put at 0.3 kg. but increases up to 3.0 kg. for foul bottoms with shells. For the "Napoléon" with slightly oxidized copper it was 0.44 kg.

49) The IHP is obtained by multiplying the resistance W by v and dividing by 75 and a coefficient η representing the efficiency of the machinery as in Eq. 192. **Determination of the IHP.**

$$IHP = \frac{Wv}{75\,\eta}$$

only that here η is, in accordance with the table on p 336, to be taken at about 0.55 for small and 0.65 for large engines.

50) It is to be remarked that the ships designed for the French Navy at the time DUPUY DE LÔME published his formula (1865) all had rather hollow entrances, which must be borne in mind in comparing angles of entrance and therefore also in determining the coefficients. As well because of this circumstance as because of the difficulty and uncertainty attending the measurement of the mean angle of entrance (§ 38, 14) and further owing to the vagueness of the coefficients, DUPUY DE LÔME'S formula was not a favourite in practice. Like BOURGOIS'S it is now pretty obsolete and is not likely to be applied in future. **Value of the Formula.**

*) E. FLACHAT. Navigation à vapeur transocéanienne.

Table of Resistance according to Dupuy de Lôme.

Name	$\mathfrak{B}$ in sqm.	$S=GL$ in sqm.	v in knots.	v in m. per sec.	W in kg.	$EHP = \frac{Wv}{75}$	IHP	$\eta = \frac{EHP}{IHP}$	Remarks
1	2	3	4	5	6	7	8	9	10
Ajaccio	13.4	362	9.80	5.037	1975	132	221	0.597	Paddle steamer with feathering floats
Narval, clean . .	21.0	458	9.00	4.626	2450	151	271	0.557	" " " common "
Nar al, foul . .	21.0	458	7.00	3.598	2450	117	228	0.513	" " " " "
Sphinx	23.0	497	8.80	4.523	2560	154	285	0.540	" " " " "
Labrador . . .	53.0	1034	10.20	5.243	6990	488	744	0.655	" " " " "
Charlemagne . .	89.0	1398	9.54	4.903	9420	575	921	0.624	Screw steamer with 2 bladed propeller
Napoléon . . .	99.0	1585	13.15	6.670	19000	1712	2602	0.658	" " " 4 " "
Napoléon . . .	100.0	1610	9.93	5.104	10000	680	1030	0.660	" " " " " "

§ 39.

Calculation of the Horse-power by the Stream-line Theory.

1) The formulæ for determining the resistance of ships by the stream-line theory may be divided into two groups viz. **Classification of the Formulæ.**

I. Formulæ which only regard the friction-resistance,
- a) RANKINE'S formula,
- b) KIRK'S formula;

II. Formulæ which regard friction, eddy, and wave-resistance,
- c) FROUDE'S formula,
- d) TIDEMAN'S formula,
- e) RAUCHFUSS'S formula.

2) **I. Formulæ which only regard friction resistance** — a. **Rankine's Formula** *) **Rankine's Formula.** is based upon his opinion that for a well-formed ship, i. e. one designed according to SCOTT RUSSEL'S rules (see § 37, 25) the eddy and wave resistance is so small that is has no perceptible effect on her progress, a view which is certainly not borne out by the experience of the present day. RANKINE therefore considers that the whole resistance is limited to the friction on the ship's skin caused by the small eddies. This eddy-resistance of RANKINE'S, which must not be confounded with FROUDE'S eddy-resistance W_v (see § 37, 23), is a combination of the direct and indirect effects of the adhesion to the ship's skin of the water particles gliding over it, which adhesion, together with the low elasticity of the surrounding water, causes the formation of a great number of small eddies on the ship's skin. The velocity with which the water particles move in these eddies bears a certain proportion to the velocity with which they glide over the skin, therefore the eddy velocity depends upon the "height due" $\frac{v^2}{2g}$ of the gliding particles. But the velocity of gliding of the particles is in a certain ratio to the ship's speed which ratio is influenced by the form and position of every element of her skin, and the height due to the velocity of gliding equals the height due to the ship's speed multiplied by the square of the above ratio. Further, the quantity of water to which the eddying motion is communicated by an element of the skin, while the ship traverses unit distance, is proportional to the area of the skin-element multiplied by the ratio of the velocity of gliding, at the element, to the ship's speed. The total resistance of the ship may therefore be expressed as follows.

*) W. J. M. RANKINE. Shipbuilding theoretical and practical. London. 1866. p. 77.

$$W = f \frac{\gamma}{2g} v^2 \int q^3 \, ds \text{ lbs. engl. } \ldots\ldots\ldots \quad (200^a)$$

where ds is the area of the element of the wetted surface, and q the ratio of the velocity of the gliding particles at the element, to the ship's speed.

Augmented Surface.

3) This integral, representing the summation of the products of the areas of the skin-elements and the cubes of the ratios of their respective speeds of gliding to the ship's speed, is called by RANKINE the *augmented surface*. He chose the expression *augmented surface* in order to shew that he did not regard the friction-resistance as proportional to the wetted surface, but that the variability of the velocity of gliding of the particles at various points of the skin was taken into account. (See also ISHERWOOD's Formula § 38, 36).

Abbreviated calculation of the augmented surface.

4) The exact computation of the augmented surface is so tedious and troublesome that all the requirements of practice can be satisfied by an approximate method of RANKINE'S which has shewn itself to be sufficiently accurate when compared with the actual results of ships of the most diverse sizes and types. This method is based upon the fact that the area of a trochoidal *) strip of given width (Table 9, Fig. 4) equals the length on the base AB, into the width, into a coefficient α, which

$$= 1 + 4 \sin^2\varphi + 4 \sin^4\varphi,$$

where φ is the angle between the base line AB and a tangent DE to the trochoid at the point of contrary flexure D. The augmented surface of a ship regarded as a trochoidal strip would therefore be determined as follows. AB is the length of the ship, the width is her mean girth G to waterline (calculated by SIMPSON'S RULE), the coefficient α is the mean value of all the angles taken as above for all the waterlines in the fore-body. The expression for the augmented surface is then

$$S q^3 \, ds = \alpha G L = G L (1 + 4 \sin^2\varphi + \sin^4\varphi) \text{ sq. ft. engl.}$$

Calculation of the Resistance.

5) Substituting in Eq. 200^a, v in knots $= 1.689\, v$ ft. engl. per sec., G and L in ft. engl., the coefficient of friction $f = 0.0036$ chosen by RANKINE from WEISBACH'S observations on the flow of water in iron pipes, — further $\gamma = 64$ lbs. engl., the weight of a cubic ft. of water, and g the acceleration of gravity $= 32.2$ ft. engl., we have

$$f \frac{\gamma}{2g} v^2 \quad 0.0036 \frac{64}{2 \times 32.2} \times 1.689^2 v^2 = \text{about } 0.01\, v^2$$

*) A trochoid is the profile of a rolling wave taken from crest to crest.

consequently

$$W = 0.01\, v^2 \int q^3\, ds = 0.01\, v^2\, G\, L\, (1 + 4 \sin^2\varphi + \sin^4\varphi) \text{ ℔s. engl.} \quad (200^b)$$

This expression, based upon an assumed eddy formation at the ship's skin, not only embraces the direct action of the water adhering to the wetted surface, but also its indirect action manifested as an increase of pressure at the bow (bow-wave) and diminution of pressure aft (stern-wave too far aft see § 38, 41).

6) RANKINE computes the IHP by multiplying the resistance W in ℔s. engl. by the ship's speed 101.3 v ft. engl. per min. and dividing by $33000 \times \eta$, the efficiency of the machinery, **Determination of the IHP.**

$$IHP = \frac{101.3\, W v}{33000\, \eta}.$$

RANKINE puts $\eta = 0.63$ for fairly good engines, thus, substituting the value of W he gets

$$IHP = \frac{0.01 \times 101.3}{33000 \times 0.63}\, v^3\, G\, L\, (1 + 4 \sin^2\varphi + \sin^4\varphi).$$

For the numerical coefficient which is exactly equal to $\frac{1}{20823}$ he substitutes the round number $\frac{1}{20000}$, so that for well-formed iron steamers with clean bottom

$$IHP = \frac{v^3\, G\, L}{20000}\, (1 + 4 \sin^2\varphi + \sin^4\varphi) \quad \ldots\ldots. \quad (201)$$

It is to be noted that the coefficient 20000 may be somewhat increased for coppered vessels. On the other hand it falls for ships of unsuitable form. For instance, for a vessel with too short entrance it was found to be only 19000 and for another with too short run as low as 16000 even.

7) As a proof of the correctness of his formula RANKINE compares the figures given by it with the trial-trip results of various ships of the British Navy, such as the "Warrior", "Fairy", and "Victoria & Albert". Although this formula can only be used when the exact drawing of the ship is available and the calculation even then is very tedious and inaccurate on account of the measuring of the angles of entrance, still in its time it has been much employed in practice. Experience has shewn however that the figures arrived at by its means are only applicable to large ships of fine form; for other vessels they are generally too low. **Value of the Formula.**

8) b. **Kirk's Formula** *) is really only a simplification of RANKINE'S sug- **Kirk's Formula.**

*) A. C. KIRK. On a method of analysing the forms of ships etc. Transactions of the Instutition of Naval Architects. London 1880. p. 96.

gested by the late WILLIAM DENNY. KIRK avoids the laborious calculation of the rubbing (or wetted) surface by constructing a solid — *a block-model*, the surface of which represents very closely the wetted surface of the ship. The process of constructing this block-model and the method of calculating the wetted surface from it is as follows.

Dimensions of the block-model.

9) Calling

L the ship's length in m.

T her mean draught in m.

l the length of the entrance and run in m.,

we have by Fig. 5 Plate 9

$\frac{l}{2} B T$ the volume of the entrance or of the run in cb. m.

$(L-2l) B T$ „ „ „ „ middle body in cb. m.

$(L-l) B T$ „ „ „ „ whole immersed part of the ship $=$ the displacement in cb. m. $= D$;

and if $B T$ represents the area if immersed midship section in sq. m $= \mathfrak{M}$, it follows that

$$L - l = \frac{D}{\mathfrak{M}}$$

and thus the length of the entrance and run

$$l = L - \frac{D}{\mathfrak{M}} \text{ m. is determined.}$$

The other dimension to be decided on, B, comes out

$$B = \frac{\mathfrak{M}}{T} \text{ m.}$$

Surfaces of the block-model.

10) The surfaces of the sides and bottom of the model give the wetted surface, thus

the bottom, $(L - 2l) B + l B = (L - l) B$ sq. m.

the sides of the middle body $2 (L - 2l) T$ sq. m.

„ „ „ „ entrance and run $4 T \sqrt{l^2 + {}^1/_2 B^2}$ sq. m.

The wetted surface is therefore

$$O = (L - l) B + 2 T \left[(L - 2l) + 2 \sqrt{l^2 + {}^1/_2 B^2}\right] \text{ sq. m.} \quad (202)$$

The surface, thus calculated, always comes out rather greater than the actual wetted surface of the ship and is therefore, according to KIRK'S experience to be reduced by

about 2 % for particularly full ships,

3 „ „ ordinary steamers,

5 „ „ fine-lined steamers with not too hollow water-lines,

8 „ „ very fine steamers with very hollow water-lines.

Proportions of the block-models.

11) By Fig. 5 Plate 9

$$\frac{{}^1/_2 B}{2} = \tan \frac{\alpha}{2}$$

from which the angle of entrance α is determined. Experience hitherto gained gives the proportions of the block-models for different types of ships as stated in the following table

Table of the Proportions of Block-models.

Type of Ship	Speed in knots	Angle of entrance α in degrees	Length of entrance l
Large fast ocean-steamers	14 and above	18 to 15	$0.30\,L$ to $0.36\,L$
Ordinary large ocean-steamers	12 to 14	21 „ 18	$0.26\,L$ „ $0.30\,L$
Ordinary cargo-steamers	10 „ 12	30 „ 22	$0.22\,L$ „ $0.26\,L$

The length L is taken from the fore side of stem to the after side of the body-post. For comparing screw steamers with each other however the length L can be regarded as between perpendiculars, i. e. to the after side of rudder-post, as the other length is generally unknown. But in comparing small and middle-sized screw steamers with paddle steamers, it is advisable to measure the length of the former between perps. neglecting the screw aperture.

12) KIRK calculates the IHP from the wetted surface Eq. (202), and like RANKINE, puts the resistance at **Determination of the IHP.**

$$W = 0.01\,v^2\,S \text{ lbs. engl.}$$

If we substitute $v = 10$ knots, we get

$$W = 0.01 \times 100 \times S = S \text{ lbs. engl.}$$

which expresses RANKINE'S Rule

"The resistance at 10 knots is 1 lb. per sq. ft. and varies as the speed squared"

From this of course the EHP may be expressed as usual

$$EHP = \frac{W \times 1.689\,v}{550} = \frac{0.01\,v^2\,S \times 1.689\,v}{550}.$$

and inserting RANKINE'S efficiency η

$$IHP = \frac{W \times 1.689\,v}{550 \times \eta} = \frac{0.01\,v^2\,S \times 1.689\,v}{550 \times \eta} = \frac{0.01 \times 1.689\,S\,v^3}{550 \times 0.63}.$$

Putting $S = 100$ sq. ft. and $v = 10$ knots, we have

$$IHP = \frac{0.01 \times 1.689 \times 100 \times 1000}{550 \times 0.63} = 4.874 \text{ or about } 5$$

or KIRK'S Rule, viz.

"To driwe an ordinary steamer 10 knots requires 5 IHP per 100 sq. ft. of her block-model surface, and the power varies as the speed cubed"

13) According to KIRK therefore **Determination of the Coefficients.**

$$IHP = \frac{v^3}{10^3} \times \frac{S}{100} \times c = \frac{c\,S\,v^3}{100000} \quad \ldots\ldots\ldots \quad (203)$$

Subsituting S in sq. ft. engl. and v in knots,

$c = 4$ for ships with very fine form, perfectly smooth and clean skin, and very efficient engines.

$c = 5$ for ships whose block-model about corresponds with the table on p. 341.

$c = 6$ to 7 for heavy beamy war-ships.

Substituting S in sq. m. in this formula, we can take c horse-power per 10 sq. m. of surface. The error which is thus involved (100 sq. ft. engl. are only 9.290 sq. m.) is negligible in comparison with that of assuming that the power due to the resistance varies as the speed cubed, a law which according to all experience hitherto, is only very approximately correct. To assist comparison, the coefficient c in the tables on p. 348 to p. 352 is computed per 10 sq. m. of block-model surface and its values for 10 knots speed are distinguished by thick figures.

Value of the Formula.

14) Since the younger FROUDE and the late WILLIAM DENNY*) have applied the results obtained from their model-trials, corroborated later by progressive trials of the steamers built from the models, to the improvement and development of KIRK'S method, it has become in England the one almost exclusively employed. As continued exertions are being made to fix, with greater precision and for various types of ships, the difference between the actual wetted surface and that of the block-model, as well as more accurate and gradated values for the coefficient c, it is to be expected that KIRK'S process, supported as it is by the experience of the first English authorities, will also be welcomed beyond England as its great simplicity entitles it to be. Tables of the results obtained by the two FROUDES and WILLIAM DENNY are given on p. 345 to p. 352. The first table contains the dimensions of the ships and the block-models as well as several important ratios. The second table gives the *IHP* determined on the progressive trials for the various speeds, also the coefficient c shewing the respective *IHP* per 10 sq. m. of block-model surface. In recent years particularly exact trials have been made with the new ships of the United States Navy, the official results of which have been published with laudable accuracy**). For the latest ships the wetted (not the block-model) surface has been calculated, also the *IHP* per 100 sq. ft. engl., here taken as = 10 sq. m. in order to conform with the rest of the tables. From the actually observed

*) W. DENNY. On the speed and carrying of screw-steamers. Engineering 1882. I. pp. 295, 396 and 537.

**) Journal of the american society of naval engineers. Washington 1892, 1893, 1894.

power at the higher speeds attained, the *IHP* per 100 sq. ft. (= 10 sq. m.) of wetted surface at 10 knots has been calculated upon the assumption that it varies as the 3.5 th. power of the speed. These very useful values are given in the following table.

Table of Results of recent American War-ships.

	1st-class Cruiser "Columbia"	2nd class Cruiser "New York"	3nd class Cruiser "Marblehead"	Gunboat "Machias"	Gunboat "Castine"	Coast defence ship "Monterey"	Training ship "Bancroft"	Torpedo-boat "Cushing"
1	2	3	4	5	6	7	8	9
Length on water-line in m.	125.45	115.82	78.33	57.91	57.91	78.03	57 15	42 06
Beam on water-line in m.	17.73	19.58	11.28	9 75	9.75	18.00	9.75	4 34
Mean trial-trip draught in m.	6.83	7.28	4.38	3.58	3.58	4.39	3.48	1.48
Displacement on trial trip in tons	7350 00	8480.00	2054.00	1067.50	1067 50	4000 00	832 00	105.30
Immersed midship-section on trial trip in sq. m.	104.05	125.41	43 10	30 16	30.16	71.72	25.73	4.44
Wetted surface on trial-trip in sq. m.	2563.06	2862.7	1040 12	633.20	633.20	1783.98	571.33	167.68
Mean indicated horse-power	17991.11	16947.29	4863.30	1794.49	2127 93	4986 87	1183.34	1754.45
IHP per 10 sq. m. wetted surface at highest speed attained	67.09	56.47	46.76	27.48	32.13	27.31	19.24	97.20
IHP per 10 sq. m. wetted surface at 10 knots speed	3.50	4.21	7.50	5 97	6.16	10 73	5.40	8.53
Mean speed on trial trip in knots	22 80	21.00	18 44	15.46	16.03	13 60	14 37	22.48

15) **II. Formulæ which regard friction, eddy, and wave-resistance.** c. **Froude's Formula*)** does not give the resistance of the ship directly, but only the engine-power indirectly. FROUDE arrived at this formula chiefly from his trials with completed steamers and towing experiments with models. His best-known trial is that made at the instance of the British Admiralty in August and September 1871 at Portsmouth with the sloop "Greyhound" of 1160 tons displacement. She was towed by the corvette "Volage", a dynamometer being inserted in the tow-rope, giving directly the resistance of the towed vessel, which came out as follows **Froude's Formula.**

at 4 knots, 0.6 tons engl. = 610 kg.
6 ,, 1.4 ,, ,, = 1412 ,,
8 ,, 2.5 ,, ,, = 2540 ,,
10 ,, 4.7 ,, ,, = 4775 ,,
12 ,, 9.0 ,, ,, = 9144 ,,

16) This resistance FROUDE analysed as described in § 37, 21, but did not venture to base upon it a formula for predicting the resistance of all kinds of ships, because, as he himself stated he was not in a position to judge how far the "Greyhound" trials would suffice for the foundation of such **Determination of the *IHP*.**

*) W. FROUDE. On the ratio of indicated to effective horse power. Transactions of the Institution of Naval Architects. London 1876. p. 11. 167.

a formula or for the determination of the coefficients. But from these trials and from the trial-trip results of the "Merkara", "Taupo", and "Hawea" (see the table pp. 345 to 352) as well as those of the "Arbutus" and "Pachumba" he developed his method of calculating the IHP, (§ 30, 18 to 21) expressed by the formula

$$IHP = 2.315\, EHP + 0.385 \frac{v}{V} EHP \quad \ldots \ldots \ldots \quad (204)$$

Value of the Formula.

17) FROUDE specially points out in his paper that he does not put forward this formula as being a perfect one, but only as a step in the right direction. The EHP is

$$EHP = \frac{Wv}{33000}$$

for W in ℔s. and v in ft. engl. per minute. The resistance W is found from towing experiments with a model, as described below.

Experimental tanks.

18) **The experiments with models** *) were first carried out by the elder FROUDE in 1872 at Torquay in a tank built by the British Admiralty and are still continued by his son R. E. FROUDE. The late Chief Constructor of the Dutch Navy Dr. TIDEMAN also made some very notable experiments, referred to later on, in a tank at the Royal Dockyard at Amsterdam. Similar resistance trials have also been made in France by naval engineer RISBEC at the Brest Dockyard and in Italy at the Castellamare Dockyard, and it is to be hoped that the time is no longer distant when the German Navy will possess a similar experimental station. Of private firms, besides that of Mess^rs. DENNY at Dumbarton, the Uebigau shipbuilding yard near Dresden belonging to the "Kette" Elbe Navigation Co. is being fitted with a tank for this purpose.

Preparation of the models.

19) *The ship models* used in these experiments are from 4 to 6 metres long, 20 to 30 mm. thick and made of parafin wax. This material was chosen because it is easily melted so that it can be cast to a shape approaching that of the model without difficulty, does not absorb water so that it does not swell, and can be used over and over again with scarcely any loss.

The fluid parafin is poured into a clay mould, the core being filled with water to withstand the pressure of the parafin. When the cast is cold the core is taken out and the model forced out of the mould by water pressure. It is then placed on a kind of copying machine which cuts out the several waterlines exactly according to the drawing by two revolving

*) Speed experiments with ships' models. The marine engineer. 1883—1884. P. 328.

Table of the Proportions of the Block-models &c. of 41 Screw-steamers.

		Dimensions	Byculla	Colaba	Chilka I. run	Chilka II. run	Sirdhana	Scindia	Pau Tah I. run	Pau Tah II. run	Fung Shun	Wacatipu	Shunlee I. run	Shunlee II. run	Hawea I. run	Hawea II. run	Goa	Nemesis
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
Hull	1	Length on Water-line m.	80.466	80.466	86.866	86.866	94.486	94.486	68.579	68.579	68.579	88.390	70.103	70.103	65.531	65.531	90.524	103.630
	2	Breadth extreme "	9.753	9.753	10.668	10.668	11.887	11.887	10.363	10.363	10.363	10.058	10.668	10.668	8.229	8.229	10.058	12.497
	3	Depth of hold "	8.534	8.534	8.077	8.077	8.382	8.382	6.401	6.401	6.401	7.620	8.763	8.763	4.461	4.461	8.077	—
	4	Trial-trip mean draugth "	3.543	3.543	3.975	3.975	5.708	4.775	2.565	3.759	3.569	4.540	2.756	4.089	3.492	3.962	4.470	4.851
	5	Depth of Keel cm.	24.130	24.130	25.400	25.400	25.400	25.400	3.175	3.175	3.175	3.175	2.857	2.857	19.050	19.050	25.400	—
	6	Mean construction draugth ex Keel m.	3.301	3.301	3.718	6.120	5.452	4.520	2.533	3.727	3.535	4.496	2.896	4.060	3.301	3.770	4.215	4.258
	7	Displacement tons	1807	1807	2367	4264	4584	3676	1175	1859	1748	2784	1321	2129	1105	1301	2502	2916
	8	Area of Midshipsect. sq. m.	29.356	29.356	37.067	62.614	61.778	50.630	34.001	37.067	35.116	42.734	27.405	41.712	25.083	28.985	39.482	43.198
Block-model	9	Length of entrance and run $= l$ in m.	20.451	20.451	24.566	20.441	22.127	23.652	22.219	19.659	20.024	24.840	23.103	20.330	22.555	21.792	28.711	37.795
	10	Half breadth of midl. body $^1/_2 B$ m.	4.450	4.450	4.983	5.120	5.669	5.599	4.877	49.68	4.968	4.755	5.023	5.136	3.779	3.840	4.681	4.956
	11	Length of the middle-body $L - l$ m.	39.562	39.562	37.733	45.992	50.229	471.82	24.139	29.260	28.528	38.709	23.895	29.442	20.421	21.945	33.100	28.041
	12	Length of entrance and run at side $= \sqrt{l^2 + {}^1/_4 B^2}$ m.	20.909	20.909	25.054	21.061	22.860	24.291	22.737	20.268	20.634	25.298	236.52	20.969	22.860	22.127	29.077	38.129
	13	Half angle of entrance $= {}^1/_2 \alpha$	12° 17'	12° 17'	11° 28'	14° 4''	14° 22'	13° 19'	12° 23'	14° 11'	13° 56'	10° 50'	12° 16'	14° 11'	9° 31'	10° 0'	9° 16'	7° 29'
	14	Surface of bottom sq. m.	534.082	534.082	620.956	680.310	820.400	793.180	452.145	486.053	482.337	603.943	472.118	511.230	324.871	335.926	578.470	652.530
	15	" " sides "	537.240	537.240	654.490	1078.569	1046.983	866.014	352.650	520.240	493.490	802.842	388.415	579.603	436.630	499.250	583.790	909.301
	16	" " total "	1071.323	1071.323	1274.216	1758.875	1867.290	1659.294	804.793	1006.300	975.821	1406.770	860.514	1090.832	761.510	835.170	1353.925	1561.835
Propeller	17	Propeller immersed or emerged cm.	out 34.289	out 26.670	out 38.099	in 134.620	in 46.989	out 7.620	out 97.788	in 6.350	in 11.430	in 1.270	out 83.818	in 27.939	in 19.050	in 26.670	out 22.860	out 43.179
	18	Propeller Disc area } total sq. m.	16.443	16.443	18.673	18.673	21.088	21.088	9.568	9.568	9.568	16.443	11.427	11.427	10.498	10.498	17.558	21.460
	19	Propeller Disc area } immersed "	15.793	16.072	18.023	18.673	21.088	20.995	8.268	9.568	9.568	164.43	9.568	11.427	10.321	10.498	17.186	20.717
	20	Immersed propeller disc area / one-tenth of total block-model surface	0.147	0.151	0.142	0.106	0.113	0.126	0.103	0.095	0.098	0.117	0.112	0.105	0.135	0.125	0.127	0.133

Table of the Proportions of the Block-models &c. of 41 Screw-steamers.

		Dimensions	Chanda	Ettore	Assam	Africa	Taupo	Glasgow	General Peel	Merkara	Afrika	Pretoria	German	Manora I. run	Manora II. run	Rotomahana I. run	Rotomahana II. run	Iris
			20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35
Hull	1	Length on Water-line m.	96.010	93.572	83.666	96.010	655.31	59.435	77.723	112.470	79.247	106.680	106.680	115.820	115.820	86.866	86.866	91.438
	2	Breadth extreme ,,	10.058	10.058	10.363	10.058	82.29	8.890	9.601	11.277	10.058	12.192	11.887	11.582	11.582	10.688	10.688	13.995
	3	Depth of hold ,,	8.077	7.925	7.925	8.077	4.461	5.486	7.848	8.991	7.925	9.906	9.449	9.300	9.300	7.620	7.620	8.255
	4	Trial-trip mean draugth ,,	4.502	4.153	4.712	4.292	3.492	4.013	3.962	49.53	4.572	5.778	5.708	4.381	5.010	4.585	4.585	5.512
	5	Depth of Keel cm.	25.400	25.400	22.860	25.400	19.050	—	21.590	27.939	22.860	27.939	24.130	27.939	27.939	3.492	3.492	27.939
	6	Mean construction draught ex Keel m	4.249	3.898	4.480	4.058	3.301	3.718	3.746	4.672	4.343	5.498	5.468	4.102	4.730	4.530	4.773	5.233
	7	Displacement tons	2738	2348	2375	2571	1105	1083	1715	4003	2002	4662	4613	3588	4278	2467	2698	3343
	8	Area of Midshipsect. s. qm.	39.761	36.881	40.226	37.624	25.083	28.056	32.143	48.772	39.575	61.871	60.013	43.013	50.259	44.406	46.450	9.290
Block-model	9	Length of entrance and run $= l$ in m.	28.833	31.455	260.61	29.381	22.555	21.792	25.663	32.399	29.899	33.223	31.699	34.472	32.796	32.734	32.613	41.299
	10	Half breadth of midl. body $^1/_2 B$ m.	4.678	4.724	44.89	4.657	3.779	3.773	4.288	5.218	4.557	5.625	1.672	5.245	5.312	4.877	4.907	6.202
	11	Length of the middle-body $L - l$ m.	38.343	30.631	31.546	37.246	20.421	15.849	26.394	47.669	19.415	40.233	43.281	46.877	50.229	21.396	21.640	8.839
	12	Length of entrance and run at side $= \sqrt{l^2 + {^1/_4} B^2}$ m.	29.199	31.789	26.456	29.748	22.860	22.127	26.029	32.826	30.235	33.679	32.155	34.868	33.223	33.100	32.948	41.757
	13	Half angle of entrance $= {^1/_2} \alpha$	9° 13'	8° 33'	9° 46'	9° 0'	9° 31'	9° 49'	9° 29'	9° 9'	8° 38'	9° 37'	9° 49'	8° 39'	9° 12'	8° 28'	8° 33'	8° 33'
	14	Surface of bottom sq. m.	628.561	568.664	517.453	620.572	324.871	283.902	446.478	835.728	450.750	826.800	823.093	853.930	882.550	528.230	532.410	621.965
	15	,, ,, sides ,,	822.072	734.559	757.135	781.382	436.630	447.025	588.243	1059.060	693.963	1183.545	1177.043	955.010	1103.626	797.268	828.668	968.300
	16	,, ,, total ,,	1450.668	1321.224	1274.590	1401.954	761.501	730.940	1034.900	1894.800	1200.453	2010.356	2000.137	1808.760	1986.156	1325.500	1361.078	1590.261
Propeller	17	Propeller immersed or emerged cm.	out 10.160	out 76.199	out 38.099	out 17.780	out 20.320	out 41.909	out 10.160	in 10.760	out 7.620	in 24.130	in 17.780	out 76.199	out 12.700	in 44.449	in 24.130	in 85.088
	18	Propeller Disc area } total sq. m.	17.558	17.558	19.230	17.558	10.498	12.356	13.285	19.881	17.558	22.296	22.296	22.296	22.296	15.328	15.328	23.968
	19	Propeller Disc area } immersed ,,	17.465	15.700	18.580	17.279	10.219	11.612	13.192	19.881	17.465	22.296	22.296	20.345	22.203	15.328	15.328	23.968
	20	$\frac{\text{Immersed propeller disc area}}{\text{one-tenth of total block-model surface}}$	0.120	0.119	0.146	0.123	0.134	0.159	0.127	0.110	0.145	0.119	0.111	0.112	0.112	0.116	0.112	0.151

		Dimensions	Te-Anau	Buenos Ayrean	Booldana	Ravenna	Bhundara	Bancoora	Cathay	Venetia	Quetta	Kangra	City of Rome I. run	City of Rome II. run	Aries	Normandie	Monarch	Tacapura
			36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51
Hull	1	Length on Water-line m	82.295	117.350	97.534	115.820	97.534	97.534	109.730	106.680	115.820	86.866	165.20	165.35	42.212	140.000	45.642	67.055
	2	Breadth extreme „	10.363	12.801	12.192	12.192	12.192	12.192	11.887	11.582	12.192	10.668	15.849	15.849	6.096	15.214	6.858	9.753
	3	Depth of hold „	7.619	8.254	8.534	8.534	8.686	8.686	9.296	8.763	9.296	8.077	11.811	11.811	3.632	11.404	4.114	5.638
	4	Trial-trip mean draugth „	5.003	5.125	5.557	5.410	5.207	5.156	4.991	5.461	5.658	4.419	6.807	6.858	2.807	6.339	3.016	3.181
	5	Depth of Keel cm.	4.444	27.939	25.400	27.939	25.400	25.400	30.479	30.479	27 939	25.400	31.749	31.749	15.000	30.479	15.000	—
	6	Mean construction draught ex Keel m.	4.940	4.846	5.303	5.123	4.952	4.907	4.680	4.907	5.379	4.170	6.493	6.553	2.654	6.038	3.016	3.181
	7	Displacement tons	2755	4820	4720	4855	4370	4340	3639	4000	5050	2655	11150	11230	295	7975	369	1225
	8	Area of Midshipsect. sq. m.	49.330	58.805	61.658	59.456	57.319	56.669	48.772	54.439	61.314	41.805	95.222	96.609	10.284	82.867	12.541	28.613
Block-model	9	Length of entrance and run $= l$ in m.	26.944	36.016	21.701	34.797	21.976	21.640	35.871	33.860	34.197	23.926	49.306	49.894	13.786	44.625	16.484	24.619
	10	Half breadth of midl. body ${}^1/_2 B$ m.	4.968	6.065	5.821	5.791	5.791	5.790	5.212	5.550	5.699	5.006	7.336	7.385	1.937	6.857	2.189	4.482
	11	Length of the middle-body $L - l$ m.	28.407	45.11c	54.029	46.024	55.739	54.253	38.005	38.949	47.423	39.014	66.750	67.055	14.642	50.751	12.674	17.806
	12	Length of entrance and run at side $= \sqrt{l^2 + {}^1/_4 B^2}$ m	27.401	36.635	22.463	35.416	22.738	22.402	36.301	34.317	34.652	24.445	50.185	49.649	13.921	45.149	19.705	25.377
	13	Half angle of entrance $= {}^1/_2 \alpha$	10° 28'	9° 32'	15° 0'	9° 26'	14° 45'	14° 57'	8° 44'	9° 19'	9° 28'	11° 50'	8° 28'	8° 29'	8° 0'	8° 44'	7° 34'	10° 21
	14	Surface of bottom sq. m.	550.90	985.85	881.99	938.29	874.19	876.97	768.55	808.23	930.86	631.52	1702.79	1701.58	110.08	1308.87	127.64	381.63
	15	„ „ sides „	817.52	1190.00	1050.45	1198.40	981.02	970.80	1036.31	1055.37	1255.96	732.51	2260.94	2271.76	238.47	1788.71	296.93	431.79
	16	„ „ total „	1368.42	2175.85	1932.44	2136.69	1855.20	1847.77	1804.86	1863.60	2186.82	1364.04	3963.73	3973.34	348.56	3097.58	424.57	813.42
Propeller			in	out	in	in	in	in	out	in	in	out	out	out	in	out	in	out
	17	Propeller immersed or emerged cm.	105.4	27.9	31.7	22.8	27.9	22.8	73.6	6.3	24.1	31.7	16.5	26.6	53.3	19.0	30.5	34.6
	18	Propeller Disc area total sq. m.	12.356	23.597	21.088	23.597	21.088	21.088	22.946	21.739	22.389	18.673	41.968	41.968	4.102	35.256	7.320	10.810
	19	Propeller Disc area immersed „	12.356	23.220	21.088	23.597	21.088	21.088	22.088	21.739	22.389	18.115	41.805	40.226	4.102	35.032	7.320	10.305
	20	Immersed propeller disc area / one-tenth of total block-model surface	0.090	0.109	0.109	0.110	0.113	0.114	0.117	0.117	0.102	0.133	0.105	0.101	0.117	0.113	0.162	0.126

Table of the Progressive

№	Speed in Knots	Byculla		Colaba		Chilka I. run		Chilka II. run		Sirdhana		Scindia		Pau Tah I. run	
1	2	3		4		5		6		7		8		9	
		IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c
1	5	95	0.887	78	0.728	105	0.824	—	—	119	0.637	116	0.699	97	1.205
2	6	139	1.297	131	1.222	160	1.255	197	1.120	181	0.969	185	1.115	153	1.901
3	7	194	1.811	198	1.848	231	1.813	308	1.751	272	1.457	283	1.705	230	2.855
4	8	267	2.492	282	2.632	326	2.559	452	2.573	396	2.131	412	1.972	335	4.162
5	8.5	312	2.912	336	3.136	385	3.021	541	3.076	496	2.656	495	2.983	403	5.007
6	9	368	3.155	386	3.622	455	3.570	645	3.667	598	3.202	591	3.562	482	5.989
7	9.5	434	4.051	453	4.228	541	4.246	765	4.349	755	4.043	700	4.219	580	7.207
8	**10**	**512**	**4.779**	**525**	**4.900**	**640**	**5.028**	**912**	**5.185**	**805**	**4.311**	**830**	**5.003**	**678**	**8.424**
9	10.5	609	5.068	614	5.731	757	5.941	1082	6.152	1090	5.837	980	5.906	814	10.114
10	11	719	6.618	713	6.655	885	6.945	1273	7.238	1280	6.855	1152	6.943	950	11.804
11	11.5	848	7.850	834	7.784	1033	8.107	—	—	1460	7.819	1345	8.106	1131	14.053
12	12	—	—	—	—	1188	9.323	—	—	1665	8.917	1580	9.522	1350	16.774
13	12.5	—	—	—	—	1352	10.160	—	—	—	—	—	—	1628	20.230
14	13														
15	13.5														
16	14														
17	14.5														
18	15														
19	15.5														
20	16														
21	16.5														
22	17														
23	17.5														
24	18														
25	Highest speed	10.68		10.78		11.96		10.39		11.34		11.56		11.79	
26	Greatest IHP developed at the same	649		669		1173		1034		1382		1367		1257	

trials of the 41 Screw-steamers.

Pau Tah II. run		Fung Shun		Wacatipu		Shunleo I. run		Shunlee II. run		Hawea I. run		Hawea II. run		Goa		Nemesis	
10		11		12		13		14		15		16		17		18	
IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c
100	0.993	—	—	114	0.810	83	0.964	91	0.834	95	1.247	—	—	—	—	213	1.364
153	1.520	—	—	175	1.243	127	1.476	136	1.247	132	1.733	—	—	—	—	287	1.837
227	2.255	255	2.613	261	1.855	185	2.149	200	1.833	180	2.364	—	—	—	—	380	2.433
326	3.239	354	3.628	379	2.693	263	3.056	290	2.659	243	3.191	235	2.813	345	2.548	498	3.188
387	3.845	405	4.150	450	3.198	312	3.626	355	3.254	290	3.939	280	3.352	400	2.954	566	3.620
473	4.700	485	4.970	531	3.774	372	4.321	434	3.979	325	4.268	331	3.963	466	3.442	650	4.162
570	5.664	555	5.674	625	4.372	450	5.229	531	4.867	381	5.003	395	4.729	535	3.951	742	4.750
690	**6.857**	**667**	**6.834**	**736**	**5.232**	**548**	**6.368**	**660**	**6.050**	**438**	**6.355**	**472**	**5.651**	**625**	**4.616**	**859**	**5.499**
835	8.298	790	8.096	852	6.156	675	7.825	825	7.563	516	6.776	565	6.765	717	5.296	993	6.352
1007	10.007	941	9.643	1026	7.293	833	9.680	1021	9.360	612	8.037	681	8.154	844	6.234	1152	7.375
1222	12.143	1158	11.867	1220	8.672	1037	11.934	1270	11.642	735	9.652	820	9.818	958	7.076	1346	8.618
1480	14.707	1454	15.247	1444	10.265	1300	15.107	1613	14.787	912	11.976	992	11.878	1126	8.317	1592	10.193
1790	17.788	1907	19.543	1712	12.169	1642	19.199	2065	18.930	1140	15.023	1214	14.536	1302	9.676	1900	12.165
11.61		11.67		11.76		12.30		11.85		11.86		11.86		11.94		12.08	
1275		1243		1325		1498		1488		854		855		1103		1641	

№	Speed in Knots	Chanda		Ettore		Assam		Africa		Taupo		Glasgow		General Peel	
		19		20		21		22		23		24		25	
		IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c
1	5	130	0.896	—	—	—	—	—	—	102	13.39	89	1.217	—	—
2	6	188	1.296	—	—	—	—	—	—	145	1.904	131	1.792	—	—
3	7	264	1.820	—	—	—	—	—	—	203	2.666	186	2.544	—	—
4	8	364	2.509	292	2.250	—	—	314	2.240	276	3.624	259	3.543	284	2.744
5	8.5	428	2.950	340	2.573	—	—	367	2.618	323	4.242	302	4.131	335	3.236
6	9	504	3.474	410	3.103	—	—	429	3.060	375	4.924	350	4.787	395	3.816
7	9.5	586	4.040	477	3.610	595	4.[illegible]68	499	3.559	438	5.752	405	5.540	465	4.492
8	**10**	**692**	**4.770**	**578**	**4.259**	**686**	**5.382**	**580**	**4.137**	**510**	**6.697**	**468**	**6.400**	**540**	**5.217**
9	10.5	806	5.556	685	5.189	803	6.300	682	4.865	596	7.826	543	7.428	624	6.029
10	11	940	6.480	830	6.289	940	7.376	802	5.721	700	9.192	638	8.727	729	7.043
11	11.5	1093	7.534	1028	8.538	1088	8.536	948	6.762	815	10.902	762	10.424	849	8.202
12	12	1270	8.755	1250	9.460	1273	9.987	1125	8.024	946	12.423	928	12.694	996	9.623
13	12.5	1478	10.189	1569	11.883	1472	11.549	1340	9.558	1100	14.445	1156	15.801	1160	11.207
14	13	—	—	—	—	—	—	1591	11.348	—	—	1500	20.520	1362	13.156
15	13.5	—	—	—	—	—	—	—	—	—	—	—	—	—	—
16	14														
17	14.5														
18	15														
19	15.5														
20	16														
21	16.5														
22	17														
23	17.5														
24	18														
25	**Highest speed**	12.12		12.14		12.16		12.42		12.44		12.65		12.88	
26	**Greatest IHP developed at the same**	1314		1333		1336		1319		1083		1246		1322	

Merkara		Afrika		Pretoria		German		Manora I. run		Manora II. run		Rotoma-hana I. run		Rotoma-hana II. run		Iris	
26		27		28		29		30		31		32		33		34	
IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c
—	—	106	0.883	164	0.816	195	0.974	—	—	—	—	121	0.129	112	0.822	204	1.282
280	1.478	160	1.333	239	1.189	268	1.340	250	13.83	—	—	171	1.290	158	1.161	284	1.742
384	2.026	233	1.941	338	1.681	364	1.820	360	1.991	386	1.944	234	1.770	218	1.640	392	2.465
514	2.712	325	2.707	470	2.338	485	2.425	500	2.765	516	2.597	316	2.384	298	2.189	540	3.396
595	3.140	380	3.165	549	2.731	564	2.820	590	3.263	595	2.995	365	2.754	344	2.528	628	3.943
677	3.572	442	3.682	639	3.178	652	3.260	683	3.778	678	3.413	426	3.214	400	2.939	716	4.502
780	4.116	514	4.281	750	3.730	756	3.780	795	4.397	780	3.926	485	3.659	460	3.374	825	5.188
900	**4.749**	**592**	**4.931**	**881**	**4.882**	**876**	**4.880**	**914**	**5.055**	**880**	**4.429**	**574**	**4.330**	**537**	**3.945**	**944**	**5.936**
1034	5.456	684	5.698	1038	5.163	1025	5.125	1050	5.807	1003	5.049	665	5.017	620	4.555	1071	6.735
1194	6.306	790	6.580	1212	6.029	1182	5.910	1215	6.707	1140	5.974	780	5.884	723	5.312	1216	7.646
1365	7.203	908	7.563	1427	7.028	1376	6.880	1410	7.799	1300	6.544	920	6.941	840	6.171	1380	8.677
1559	8.227	1040	8.663	1662	8.267	1595	7.975	1630	9.016	1490	7.500	1080	8.158	977	7.178	1576	9.910
1770	9.340	1200	9.996	1941	9.655	1850	9.025	1870	10.344	1688	8.448	1250	9.430	1130	8.302	1779	11.186
1990	10.501	1400	11.662	2265	11.266	2135	10.675	2150	11.893	1930	9.715	1463	10.052	1310	9.625	2020	12.702
2240	11.820	1600	12.328	2655	13.207	2465	12.325	2455	13.580	2182	10.980	1700	12.825	1510	11.094	2285	14.356
				3120	15.519	2850	14.250	2800	15.488	2490	12.534	1958	14.872	1765	12.967	2580	16.223
				3711	18.454	2393	16.465	3185	17.617	2804	14.215	2270	17.126	2060	15.135	2922	18.374
				—	—	—	—	3650	20.190	3210	16.172	2670	20.143	2470	18.147	3300	20.751
								—	—	3608	18.165	3150	23.767	3025	22.225	3742	23.531
																4180	25.290
																4788	30.108
																5280	33.202
																6040	39.541
																6650	41.817
12.91		13.04		13.66		13.87		14.43		14.88		15.16		15.39		18.59	
1908		1414		2804		2760		3138		3115		2808		2907		7556	

№	Speed in Knots	Te-Anau		Buenos Ayrean		Booldana		Ravenna		Bhun-dara		Bancoora		Cathay	
		35		36		37		38		39		40		41	
		IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c
1	5	122	0.82	190	0.83	139	0.66	—	—	130	0.65	125	0.63	91	0.47
2	6	175	1.18	280	1.22	206	0.99	280	1.22	190	0.95	198	0.99	149	0.77
3	7	249	1.68	400	1.72	299	1.44	390	1.69	289	1.45	310	1.56	241	1.24
4	8	346	2.34	557	2.42	424	2.04	530	2.30	444	2.21	464	2.33	362	1.86
5	8.5	405	2.73	660	2.87	505	2.43	620	2.69	538	2.70	558	2.81	437	2.25
6	9	478	3.23	775	3.37	604	2.90	705	3.07	650	3.26	666	3.35	526	2.71
7	9.5	574	3.88	909	3.95	722	3.47	815	3.54	775	3.38	790	3.97	634	3.26
8	**10**	**691**	**4.66**	**1066**	**4.64**	**865**	**4.16**	**940**	**4.09**	**910**	**4.57**	**941**	**4.73**	**761**	**3.92**
9	10.5	830	5.60	1244	5.41	1047	5.03	1080	4.69	1076	5.39	1120	5.63	902	4.64
10	11	998	6.74	1450	6.31	1275	6.13	1235	5.37	1280	6.43	1350	6.79	1061	5.46
11	11.5	1186	8.01	1691	7.36	1561	7.50	1420	6.17	1562	7.82	1651	8.30	1250	6.44
12	12	1400	9.45	1960	8.53	1918	9.22	1630	7.09	1890	9.47	2009	10.10	1450	7.46
13	12.5	1640	11.07	2238	9.74	—	—	1880	8.17	—	—	—	—	1674	8.62
14	13	—	—	2525	10.99	—	—	2150	9.35	—	—	—	—	1924	9.90
15	13.5	—	—	2820	12.28	—	—	2495	10.84	—	—	—	—	2191	11.28
16	14							2880	12.52	—	—	—	—	2459	12.66
17	14.5							3333	14.50	—	—	—	—	2731	14.06
18	15							3780	16.43	—	—	—	—	—	—
19	15.5														
20	16														
21	16.5														
22	17														
23	17.5														
24	18														
25	Highest speed	12.20		13.62		11.80		14.63		11.65		11.70		13.87	
26	Greatest IHP developed at the same	1502		2899		1788		3445		1665		1796		2381	

Venetia		Quetta		Kangra		City of Rome I. run		City of Rome II. run		Aries		Normandie		Monarch		Tacapura	
42		43		44		45		46		47		48		49		50	
IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c	IHP	c
99	0.49	186	0.79	99	0.67	—	—	—	—	—	—	—	—	—	—	—	—
168	0.84	270	1.15	151	1.03	—	—	—	—	—	—	—	—	—	—	—	—
264	1.31	380	1.61	225	1.53	900	2.01	—	—	81	2.15	—	—	91	2.08	—	—
395	1.96	528	2.24	324	2.21	1175	2.75	—	—	112	2.98	—	—	143	3.28	—	—
478	2.38	621	2.64	387	2.64	1350	3.16	—	—	133	3.54	—	—	179	4.11	—	—
577	2.88	730	3.10	461	3.14	1575	3.69	—	—	159	4.23	—	—	217	4.98	—	—
692	3.45	865	3.67	548	3.73	1800	4.21	—	—	190	5.06	—	—	260	5.97	—	—
815	**4.06**	**1020**	**4.33**	**654**	**4.45**	**2075**	**4.86**	—	—	**230**	**6.13**	—	—	**305**	**7.00**	**362**	**4.13**
958	4.77	1201	5.10	788	5.36	2375	5.56	—	—	279	7.43	—	—	350	8.03	460	5.25
1117	5.57	1403	5.96	1007	6.86	2725	6.38	—	—	—	—	—	—	403	9.25	583	6.65
1300	6.48	1634	6.94	—	—	3150	7.38	—	—	—	—	—	—	478	10.97	716	8.17
1500	7.48	1885	8.01	—	—	3575	8.37	2500	5.83	—	—	2562	7.68	—	—	862	9.84
1741	8.68	2153	9.15	—	—	4040	9.46	2910	6.79	—	—	2750	8.24	—	—	1034	11.80
2000	9.97	2432	10.33	—	—	4500	10.54	3320	7.75	—	—	2950	8.54	—	—	1237	14.12
—	—	—	—	—	—	5025	11.77	3780	8.82	—	—	3240	9.71	—	—	1480	16.90
—	—	—	—	—	—	5600	13.12	4270	9.96	—	—	3593	10.77	—	—	1810	20.67
—	—	—	—	—	—	6250	14.64	4820	11.25	—	—	4050	12.14	—	—	2350	26.83
—	—	—	—	—	—	6950	16.28	5450	12.72	—	—	4562	13.68	—	—	2500	28.55
						7675	17.98	6150	14.35	—	—	5160	15.47				
						8350	19.57	6900	16.10	—	—	5780	17.33				
						8975	21.03	7750	18.09	—	—	6530	19.58				
						9650	22.61	8630	20.14								
						—	—	9790	22.85								
						—	—	11070	25.84								
12.83		13.34		10.69		16.82		18.23		10.65		16.67		11.51		14.59	
1889		2644		859		9377		11890		297		6960		478		2510	

cutters, after which it is not difficult to finish the model by hand. An illustrated description of this machine is given in Engineering *).

Method of conducting the trials.

20) To carry out the trials, a wooden frame about 3 m. long is secured to the sides of the model and connected in such a manner with a carriage running on rails, that the model is suspended freely, except that it cannot yaw. The rails are laid straight and level about 0.5 m. above the water level of the tank. A steel wire rope attached to the carriage is taken to a winding drum driven by a steam-engine, arranged so that the speed of the model can be varied between $^1/_2$ and 10 knots. Before the experiment the model is brought to the desired draught by little bags of shot laid inside it and leaden weights placed on the wooden frame.

Transmission of the pull.

21) Between the carriage and the frame an arrangement is fitted to prevent the direct communication of an accelerating or retarding force to the model, as such a force would strain the delicate dynamometer springs which measure the small resistance of the model when in uniform motion. As soon as uniform motion is attained, the pull is transmitted to the dynamometer by a rod working on a knife-edge at the lower end of the dynamometer and connected with the model at the height of the water line by means of a cord.

Dynamometer.

22) *The dynamometer* consists of two bell-cranks placed one above the other and having steel knife-edge fulcra. A helical spring fixed at one end, is attached at its other end to a knife-edge on the upper arm of the lower lever. The extension of this spring measures the resistance of the model, every variation of it is recorded by a tracing pen attached to the long arm of the upper bell-crank, so as to magnify the motion, upon a revolving paper cylinder driven by one of the axles of the carriage. The mean ordinate of the curve thus described measures the resistance of the model. A second pen moved by an electric contact closed by a train of clock-work makes a dot on the paper cylinder at every half-second and thus from the distance between several of these dots the mean speed of the model can be calculated. In order to correct this recorded speed which is affected by the oscillation and stretching of the wire tow-line, a third tracing pen is fixed beside the second one and draws a line which is interrupted at every tenth foot of the travel of the carriage. These interruptions are effected by small blocks of wood, placed next the rails and

*) Engineering II. 1886. p. 133.

spaced 10 feet apart, which come in contact with a trigger projecting beneath the bottom of the carriage.

23) *The Strophometer* is an instrument, also forming part of the whole apparatus, which serves to register any changes in the uniform motion of the model. It consists of two flat steel springs loaded with weights and attached at one end to a fly-wheel while their other ends carry a spindle moving freely through the axis of the fly-wheel. The fly-wheel is driven from one of the axles of the carriage, and the springs are deflected by centrifugal force, thus causing a movement of the spindle. This movement is recorded by a fourth tracing pen which describes a curve upon the same paper cylinder as the other three. This curve exhibits the slightest variations in the uniform speed. Finally there are two more paper cylinders attached, one at each end of the carriage and driven from one of its axles, on which two vertical rods guided in the frame and furnished with tracing pens, record any alteration of the draught at each end of the model. Strophometer.

24) From the average resistance W and the average speed v of the model as registered by the apparatus, the "corresponding" resistance $n^3 W$ for the corresponding speed $v\sqrt{n}$ of the ship (see § 36, 1) can be calculated, and of course from this corresponding resistance the *IHP* follows, upon the assumption of a certain efficiency of the machinery, say from 0.45 to 0.55. Thus, if for a vessel intended to steam $v\sqrt{n} = 13$ knots or 6.4 m per sec. a model $\frac{1}{25}$ th. of the lineal dimensions (about $\frac{1}{2}$ inch scale) is to be tried, we have Determination of the *IHP*.

$$v\sqrt{n} = 6.40 \text{ m},$$

then the speed at which the model must be towed,

$$v = \frac{6.40}{\sqrt{n}} = \frac{6.40}{\sqrt{25}} = 1.28 \text{ m}.$$

If the model's resistance $R = 0.75$ kg., the ship's resistance will be $n^3 R = 25^3 \times 0.75 = 11718.75$ kg.

and the *EHP* $11718.75 \times 6.40 = 75000.7$ mkg.

Assuming an efficiency of 0.45, the *IHP* comes out

$$\frac{75000.7}{0.45 \times 75} = 2222.25 \text{ say } 2250 \; IHP.$$

25) d. **Tideman's formula***) divides the total resistance arrived at by model-trials like FROUDE'S into the following four parts, viz. Tideman's formula.

$$W = W_r + W_v + W_w + W_p \text{ kg.} \quad \ldots\ldots\ldots \quad (205)$$

in which expression

*) B. J. TIDEMAN. Uitkomsten van proeven op den Wederstand van Scheepsmodellen. Memorial van de Marine. II. Afdeeling. pp. 75 to 90. Amsterdam 1876 to 1880.

W_r is the friction-resistance,
W_v the form-resistance, identical with FROUDE'S eddy-resistance,
W_w the wave-resistance,
W_p the resistance due to combining the effect of the negative pressure in the run occasioned by the propeller's action with that of its water-friction.

Friction-resistance. 26) *The friction-resistance* W_r TIDEMAN calculates by the following formula

$$W_r = f \gamma S v^m \text{ kg.} \qquad (206)$$

Substituting S in sq. m., the weight γ of a litre of sea-water (in TIDEMAN'S experiments $\gamma = 1.026$), and v in m. per sec., the coefficient f and the exponent m are found from the following tables according to the length of the model and of the ship and the condition of their surfaces. The tables give the results of TIDEMAN'S experiments with *clean* surfaces.

Table of Constants for smooth Paraffin Models.

Length of Model in m.	0.6	1.0	1.5	2.0	2.1	2.2	2.3	2.4	2.5	2.6	2.8	3.0	3.2	3.4	3.8	4.2	4.6	5.0	6.0
Coefficient f	0.2140	0.2025	0.1915	0.1830	0.1817	0.1805	0.1790	0.1775	0.1762	0.1750	0.1730	0.1710	0.1689	0.1669	0.1638	0.1610	0.1585	0.1565	0.15[illegible]

Table of Constants for clean surfaces.

Length of ship on water-line	Iron ships with well painted bottom		Ships sheathed with copper or zink, without projecting nailheads or wrinkles		Ships with old copper sheathing like the "Greyhound"	
in m.	f	m	f	m	f	m
5	0.1780	1.8507	0.1633	1.9015	0.2263	1.8660
10	0.1622	1.8427	0.1590	1.8525	0.2087	1.8525
20	0.1572	1.8290	0.1563	1.8270	0.1985	1.8430
30	0.1555	1.8290	0.1546	1.8270	0.1945	1.8430
40	0.1540	1.8290	0.1533	1.8270	0.1925	1.8430
50	0.1530	1.8290	0.1522	1.8270	0.1906	1.8430
60	0.1515	1.8290	0.1510	1.8270	0.1895	1.8430
70	0.1502	1.8290	0.1502	1.8270	0.1882	1.8430
80	0.1490	1.8290	0.1498	1.8270	0.1873	1.8430
90	0.1480	1.8290	0.1490	1.8270	0.1862	1.8430
100	0.1472	1.8290	0.1485	1.8270	0.1855	1.8430
110	0.1468	1.8290	0.1483	1.8270	0.1852	1.8430
120	0.1460	1.8290	0.1482	1.8270	1.1846	1.8430

Within the usual limits m is constant and $= 1.94$.

Eddy and wave-resistance. 27) *The Eddy-resistance* W_v *and the Wave-resistance* W_w are not shewn separately by the experiments, but can only be determined

together. Their determination is based upon the following rule derived from the law of corresponding speeds (§ 30, 1): if two floating bodies (a model and the ship built from it) are towed in the same fluid so that their respective speeds are to each other as the square-roots of their lineal dimensions or as the sixth roots of their displacements, i. e.

$$\frac{v}{V} = \sqrt{n} = \sqrt{\frac{l}{L}} = \sqrt[6]{\frac{d}{D}},$$

then their resistances will be to each other as the cubes of their homologous dimensions or directly as their displacements,

$$\frac{w_v + w_w + w_p}{W_v + W_w + W_p} = \frac{l^3}{L^3} = \frac{d}{D}.$$

These resistances are also proportional to the density of the fluid, and finally the dimensions of the waves caused by the motion of the bodies are proportional to the dimensions of the bodies themselves.

28) *The resistance of the model* was in general determined by TIDEMAN by a similar process to DENNY'S (see 18 to 24). TIDEMAN however departed from it to the extent of seeking to arrive at not only the total resistance but the various component resistances separately. His models were of exactly 1 metrical ton displacement = 1000 kg. and he used sea-water of 1.026 sp. g. in his tank. He first determined for different speeds v in m. per sec., the resistance of the model without propeller, viz. **Model-resistance.**

$$W_x = W_r + W_v + W_w.$$

The friction resistance W_r was subsequently calculated from the wetted surface by Eq. 206, substituting the values of f and m from the table on p. 356. Subtracting this friction-resistance from W_x, we have

$$W_x - W_r = W_v + W_w$$

as the eddy and wave-resistance. With these values of $W_v + W_w$ for the different speeds a curve (Pl. 9, Fig. 6) is constructed, having the speeds as abscissæ and their corresponding resistances as ordinates. This curve is characteristic of the combined eddy and wave-resistance of the model. TIDEMAN adopted a scale for this diagram such that in every square in Fig. 6 Pl. 9, 1 kg. of resistance corresponds to 1 dcm. per sec. speed for his models of 1000 kg. displacement in salt water.

29) TIDEMAN'S practice was to repeat the towing experiment after attaching the propeller (paddles, or single or twin screws) to the model, the propeller being actuated by special mechanism at the proper revolutions for the model's speed in each case. The resistance then arrived at is the total resistance W and **Propeller-resistance.**

the excess of W above W_x i. e. W_p is the propeller resistance. This quantity, being plotted in the above-mentioned diagram for every speed, gives the curve of the propeller-resistance of the model. TIDEMAN also devised an auxiliary diagram (Fig. 7, Pl. 9), the ordinates being displacements (for a series of vessels all built from the same model but of increasing dimensions) and the abscissæ the lengths of one knot in m. per sec. at the corresponding speeds. For instance a ship of

100	250	500	800	1000	1500	1800	2500	3000	4000	5000	6000	8000

tons displacement has in this figure an abscissa at 1 knot of

19.9	17.1	15.25	14.1	13.6	12.7	12.1	11.66	11.3	10.78	10.39	10.07	9.6mm.

By the help of these diagrams the ship's resistance can be determined in the following very convenient manner.

Ship's resistance. 30) *The resistance of the ship* at D tons displacement and V m per sec. speed is obtained by first calculating the corresponding speed of the model of 1 ton displacement by 27)

$$v = V\sqrt[6]{\frac{1}{D}}.$$

On finding the ordinate belonging to this speed in Fig. 6, Pl. 9, we have $AC =$ the resistance $W_v + W_w$ and AB the resistance W_p as every unit of length of these ordinates represents 1 kg. of resistance. These resistances which oppose the model of 1 ton displacement at speed v are the same which affect every ton of displacement of the ship at the corresponding speed V. Thus the resistances $W_v + W_w$ and W_p are determined, as it is only necessary to multiply the model's resistances by the ship's displacement. Calculating the ship's friction-resistance W_r by Eq. 206 and adding all three values, we get the ship's total resistance W as their sum.

Example. 31) *Example of Tideman's calculation of a ship's resistance.* Required the total resistance at 14 knots of the Dutch cruiser "Atjeh" of 3000 tons displacement. Fig. 6 Pl. 9 shews the resistances of the model. The abscissa $a\ b$ drawn in Fig. 7 Pl. 9 for 3000 Tons displacement is to be stepped off 14 times in Fig. 6 of the same Plate, whereupon the corresponding ordinates AC and AB give the resistances $W_v + W_w$ and W_p for every ton of displacement at 14 knots and, being multiplied by 3000, the actual resistances for the ship. On measuring the ordinates, we find

$$W_v + W_w = 1.52 \text{ kg. and } W_p = 0.86 \text{ kg.}$$

Consequently the corresponding resistances for the ship are
$W_v + W_w = 1.52 \times 3000 = 4560$ kg. $W_p = 0.86 \times 3000 = 2580$ kg.
The friction-resistance, calculated by Eq. 206, is

$$W_r = 1.026\, S\, f\, V^m.$$

For a length of ship of 81.6 m, it is found by interpolation in the table on p. 356 for an iron ship that

$$f = 0.14884 \text{ and } m = 1.829.$$

The wetted surface is 1337.5 sq. m. and the speed $V = 14$ knots $= 7.0216$ m. per sec., therefore

$$W_r = 1.026 \times 1337.5 \times 0.14884 \times 7.216^{1,829} = 7556 \text{ kg.}$$

and

$$W = 4560 + 2580 + 7556 = 14696 \text{ kg. or say } 14700 \text{ kg.}$$

32) How the *IHP* is arrived at from this resistance TIDEMAN does not say. But we can hardly go wrong by regarding the resistance $W_r + W_v + W_w$ as the ship's nett resistance of FROUDE'S formula (see § 30, 18 to 12) which has to be overcome by the effective horse-power, and further by taking the work due to the resistance W_p as 0.4 *EHP* for the "augment of resistance" and 0.04 *EHP* for the water-friction of the screw, together = 0.44 *EHP*. We then have **Determination of the IHP.**

$$EHP = \frac{(W_r + W_v + W_w)\, V}{75}$$

$$0.44\ EHP = \frac{W_p\, V}{75}$$

$$1.44\ EHP = \frac{W \times V}{75}$$

$$EHP = \frac{W \times V}{1.44 \times 75}$$

$$IHP = 2.582\ EHP = \frac{2.825 \times W \times V}{1.44 \times 75}.$$

or in the present example

$$IHP = \frac{2.582 \times 14700 \times 7.0216}{1.44 \times 75} = 2436.$$

The "Atjeh's" engines are stated to be of 2750 *IHP* and have developed a maximum of 3270 *IHP*, giving the ship a speed of 15 knots. At 14 knots she indicated 2332, which with the assumed high efficiency of 0.6, corresponds to a resistance of 14571 kg. and therefore agrees very well with the model results, as the efficiency of the engines must have been below 0.6.

33) TIDEMAN says "model experiments have in the first place a scientific value, because a sufficient number of them will furnish a means of developing a more correct theory of the motion of bodies floating in a fluid. But they have a practical value besides, in as much as they are the only means at present known of predicting with adequate accuracy the **Value of the formula.**

resistance of ships of types which diverge from any existing hitherto". The soundness of these conclusions is becoming more and more acknowledged and is not likely to be impugned today.

Rauchfuss's formula.

34) e. **Rauchfuss's formula***) divides the resistance as based on FROUDE'S "Greyhound" trials, i. e. on the stream-line theory, into four parts, viz.

$$W = W_r + W_l + W_w + W_b;$$

where W_r = the friction-resistance,
W_l = the air-resistance,
W_w = the eddy and wave-making resistance i. e. the resistance which *causes* the formation of eddies and waves,
W_b = the accompanying wave-resistance, i. e. the resistance *caused by* the eddies and waves formed by W_w.

Friction-resistance.

35) *The friction-resistance* W_r is calculated upon FROUDE'S rule, viz. that a surface more than 50 ft. engl. in length and covered with varnish or tinfoil experiences a resistance of 0.25 ℔s. engl. per sq. ft. engl. at a speed of 600 ft. engl. per min. and that this resistance varies for other speeds as the 1.83 th. power of the speed ($v^{1.83}$). Expressed on the metrical system, but for v in knots, as in all the following formulæ, this rule becomes

$$W_r = 0.0471\, S\, v^{1.83} \text{ kg.} \ldots\ldots\ldots\ldots (207)$$

Air-resistance.

36) *The air-resistance* W_l is also derived from FROUDE'S "Greyhound" trials which shewed it to be 1 ℔ per sq. ft. of thwartship projection of emerged portion of ship at a relative speed of 15 knots and to vary as the speed squared (§ 37, 28). On the metrical system therefore

$$W_l = 0.0215\, A_1\, v^2 \text{ kg.} \ldots\ldots\ldots\ldots (208)$$

Eddy and wave making resistance.

37) *The eddy and wave-making resistance* W_w represents the pressure corresponding to the resistance caused by the formation of waves and eddies. RAUCHFUSS expresses it as

$$W_w = 1.410\, v^2 \int ds \sin^3\alpha \text{ kg.} \ldots\ldots\ldots (209)$$

where $\int ds \sin^3\alpha$ signifies the magnitude and form of the wetted surface exposed to the eddy and wave-resistance and is computed by an analytical method of his own.

*) E. RAUCHFUSS. Widerstand und Maschinenleistung der Dampfschiffe. Kiel 1886.

38) *The accompanying wave-resistance* W_b is caused by the rise of level due to the eddies and waves formed by the ship; it is Accompanying wave-resistance.

$$W_b = 0.00268\, v^6 \left(\int dl \sin {}^7\alpha\right)^{\frac{1}{2}} \text{kg.} \quad \ldots \ldots \ldots (210)$$

$\int dl \sin {}^7\alpha$ is obtained by stepping off a certain length upon the run of the load water-line from the stem to midships, constructing $\sin \alpha$ for each of the spots thus obtained, raising the sine to the 7th power and summating by SIMPSON'S rule.

39) *The total resistance* W, on the metrical system and for v in knots, has the following form therefore, Total-resistance.

$$W = 0.0471\, S v^{1.83} + 0.0215\, A_1 v^2 + 1.410\, v^2 \int ds \sin {}^3\alpha + 0.00268\, v^6 \left(\int dl \sin {}^7\alpha\right)^{\frac{1}{2}} \text{kg.} \quad (211)$$

Whence follows

$$\mathfrak{H}\mathfrak{P} = \frac{W v}{75} = \frac{323\, S v^{2.83}}{1000000} + \frac{147\, A_1 v^3}{1000000} + \frac{9650\, v^3 \int dl \sin {}^3\alpha}{1000000} + \frac{18.4\, v^7 \left(\int dl \sin {}^7\alpha\right)^{\frac{1}{2}}}{1000000} \text{kg.} \quad (212)$$

40) RAUCHFUSS'S formula is open to the same objection as that raised by FROUDE in his remark quoted in 17) viz. that the coefficients are dependent upon the *one* set of trials — of the "Greyhound". If a sufficient number of coefficients, obtained from numerous towing trials of ships of various types, were available, RAUCHFUSS'S formula would certainly find more general application in practice in all cases where model experiments are not made. RAUCHFUSS'S analysis of the trial-trip results of ships of the German Navy and the useful effect of the engines as determined therefrom are already given in § 30, 23. Value of the formula.

§ 40.

Curves of Resistance.

1) DENNY*) was the first to represent graphically the results obtained on his progressive trials. His method was considerably improved upon by the elder FROUDE in 1878 and has been more and more developed since then. This graphic process consists essentially in plotting the mean speeds of the various runs as abscissæ in a system of coordinates and the corresponding horse-powers, revolutions, coal-consumptions, &c. as ordinates. A curve as fair as possible being drawn through the ends of each of these sets of ordinates then gives the horse-power &c. for any intermediate abscissa or speed. Object.

*) W. DENNY. On the trials of screw-steamships. Engineering II. 1875. p. 311.

Classification.

2) The system has been principally applied to
   a) Curves of indicated horse-power,
   b) „ „ „ thrust,
   c) „ „ slip,
   d) „ „ revolutions,
   e) „ „ cut-off,
   f) „ „ coal-consumption,
   g) „ „ coefficients of performance.

Curve of horse-power.

3) a. Curves of indicated horse-power are those which are most frequently used. Their construction is at once seen from Fig. 8, Pl. 9 in which the points *a*, *b*, *c*, and *d* represent the results of the mail-steamer "Normannia's" trials at Wemyss Bay on April 29th. 1890. The curve laid through these points passes through the origin of the coordinates, because zero speed corresponds to zero *IHP*. The study of many such curves for the most widely differing ships has shewn that at the higher speeds the work due to the resistance increases faster than with the speed cubed, as ISHERWOOD also found from his trials (see § 38, 40). The graphic method has the great advantage of affording the very best means of comparing the performances of engines and of detecting any errors and original defects at the trials.

Construction of the curve.

4) The curve of *IHP* is sometimes constructed by taking the cubes of the speeds v_1^3, v_2^3, v_3^3 as abscissæ instead of the speeds themselves, v_1, v_2, v_3. If to these abscissæ, the corresponding indicated horse-powers are drawn in as ordinates, their ends must lie in a straight line when the *IHP* varies as the speed cubed. Where this law does not apply, a deviation from the straight line will take place upwards at the higher speeds, a hump being formed, because at these speeds the *IHP* increases in a greater ratio than the cube of the speed. At lower speeds there will be a hollow because here the contrary is the case. This method has the advantage of the straight line being more easily drawn than a curve and of rendering the deviations referred to immediately apparent.

Division of the IHP.

5) W. FROUDE, as stated in § 30, 20, divided the *IHP* of a marine engine into the following six parts, viz.
   A the *IHP* actually applied to overcoming the ship's resistance,
   B the *IHP* lost through the augment of resistance due to the propeller's action on the ship's run,
   C the *IHP* absorbed by the water-friction of the screw,
   D the *IHP* required to drive the engine disconnected from the propeller,
   E the *IHP* lost by the engine-friction due to the load,
   F the *IHP* necessary to drive the pumps.

In § 30, 22 FROUDE'S opinion is given on the proportions these various parts bear to each other and Fig. 4 Pl. 10 shews the curve drawn by him shewing the division of the *IHP* for the engines of the steamer "Merkara". The portions *A, B, C, D, E* and *F* of the ordinates of this curve correspond to these separate portions of the *IHP*. Although the subdivisions may come out differently for recent engines still this diagram will always be to a certain extent a guide.

6) R. E. FROUDE*) has shewn by his model-trials (see § 39, 18) that the *IHP* curve of a marine engine only remains fair for low speeds, but that at higher speeds it exhibits certain crests and hollows which may be explained in the following manner as depending on the wave-making resistance of the ship. Every vessel moving through the water generates at her bow a wave the length of which is a function of her speed. At certain speeds the crest of this wave is situated at some point of the run (the stern-wave), and exerts a positive pressure against it, thus diminishing the ship's resistance and with it the *IHP*, so that the curve will shew a hollow in the region of this speed. If, on the other hand the ship is driven at a speed such as to form a wave of a length which brings a trough instead of a crest at her run, the resistance (see § 37, 26 and 38, 41) and the *IHP* are increased and the curve exhibits a hump. It thus appears that a wavy form of the *IHP* curve is not due, as was the universal opinion at first, to inaccurate observations or defects in the machinery, but to an unfavourable form of ship, i. e. too great a draught or an unsuitable length of entrance and run for the particular speed. These lengths must therefore be chosen in the design with due regard to the intended average speed, as given in § 37, 25. **Influence of the wave-resistance upon the *IHP*.**

7) In Fig. 7 Pl. 10 are shewn the resistance curves which a certain modern transatlantic steamer 122 m. long would have as based upon the younger FROUDE'S experiments with her model. These curves do not represent the ship's total resistance W, but the total resistance less the friction-resistance $W - W_r$, i. e. the eddy and wave-resistance $W_v + W_w$. It is plainly to be seen in what manner the ship's draught affects the resistance, as the different displacements have been arrived at by increasing the draught of one and the same model. The deeper the draught the more striking the irregularities in the (wave-making) resistance curve become. This phenomenon is due to the immersion of the **Influence of the draught upon the resistance.**

*) R. E. FROUDE. On the leading phenomena of the wave-making resistance of ships. Transactions of the Institution of Naval Architects. London 1881. p. 220.

fuller water-lines at the deeper draughts whereby the crest of the bow-wave is shifted further forward and that of the stern-wave further aft, thus either considerably reducing or increasing the resistance.

Curves of indicated thrust.

8) b. **Curves of indicated thrust** were introduced in 1876 by the elder FROUDE*) in place of DENNY'S curves of indicated horse-power. They were suggested by the generally received opinion that *IHP* cannot be regarded as a measure of propulsive effect because it leaves out of account the efficiency of the engines as well as those of the propeller and of the ship's form. But a more suitable criterion is obtained by dividing the *IHP* expressed in foot-pounds per min. by the speed of the propeller (revolutions into pitch), the quotient being of course *a force* which FROUDE called the *indicated thrust.* This quantity is what the thrust of the propeller would be if there were no losses of effect in the machinery due to friction, pumps, &c. The indicated thrust is therefore a certain multiple of the mean indicated pressure of a multiple-expansion engine reduced to the LP piston and can be calculated from the expression

$$P = \frac{p_{i_r} \times O \times 2\,H}{h} = \frac{60 \times 75 \times IHP}{h\,N} \text{ kg.} \quad \ldots\ldots \quad (213)$$

where p_{i_r} is the reduced mean indicated pressure in kg. per sq. cm. (see § 19, 15 and § 30, 12)

H the piston-stroke in m.,

O the effective area of *L P* piston in sq. cm. (see § 30, 7),

h the pitch of the screw in m.,

N the revolutions per minute.

Division of the indicated thrust.

9) The indicated thrust may be divided like the *IHP* into the following component parts, due to

I the resistance of the ship,

II the negative pressure on the run caused by the propeller.

III the water-friction of the propeller,

IV the friction of the engine disconnected,

V the friction of the engine due to the load,

VI the work of the pumps.

Of these items, II, III and IV are in FROUDE'S opinion probably proportional to the ship's resistance. VI will be approximately proportional to the revolutions squared and therefore also, at any rate for low speeds, to the ship's resistance. V is probably constant at all speeds and is so treated by FROUDE who calls it the *initial friction.* After separating out this constant quantity,

*) W. FROUDE. On the ratio of indicated to effective horse-power. Transactions of the Institution of Naval Architects. London 1876. p. 167.

the residue of the indicated thrust will be proportional to the ship's resistance, as his investigations proved.

10) On constructing a curve with indicated thrusts calculated according to Eq. 213 as ordinates, which is done for the "Merkara" in Fig. 1 Pl. 10, it will be seen that the curve does not tend, at the lower speeds, towards the origin O, but cuts the axis of y at a certain height above that point. As at zero speed of ship there is no thrust due to her resistance, the ordinate of the curve at the y axis can mean nothing else but *the constant initial friction.* If this ordinate can be determined accurately, a straight-line drawn through the end of it parallel to the x axis, will cut all the ordinates of the curve so that the constant portion of them below this parallel will be the initial friction and the parts above the parallel will be approximately proportional to the ship's resistance. If THURSTON is right in saying that in good engines there is no initial friction at all (§ 30, 30) the point A must always coincide with O. **Initial friction.**

11) The ordinate OA may be determined in the following manner. Assuming that the indicated thrust for several low speeds is known, for instance for 3, 4 and 5 knots, and making use of the fact ascertained by FROUDE that for tolerably well-formed ships of not too unusual dimensions, the ship's resistance at such low speeds consists almost entirely of friction-resistance and that this varies as the 1.87th. power of the speed, we can without any perceptible error regard the total resistance (at low speeds) as varying in the same ratio. Upon these assumptions the lower and of the curve can be replaced by a parabola whose ordinates are the 1.87 th. powers of the abscissæ, the above parallel being taken as the x axis, and this parabola will run fair into the indicated thrust curve. The construction of the parabola is very simple. At the point p (Fig. 2, Pl. 10) near the lower end of the thrust curve, draw the tangent $p'p''$, divide Oh so that $Oh' : hh' :: 0.87 : 1$, i. e. so that $Oh = 1.87\, Oh'$, erect a perpendicular at h', draw through the point of intersection B of this perpendicular and the tangent, a straight line parallel to the x axis, then the point A in which this parallel cuts the y axis, is the vertex of the parabola and OA the ordinate representing the initial friction. For ordinary comparisons in practice, especially at sea, it is sufficiently accurate to bisect the abscissa Oh in h', instead of dividing it in the ratio of $0.87 : 1$. **Usual construction of the curve.**

12) The application of these thrust curves, so easily constructed, is particularly to be recommended to all responsible engineers who are auxious to know exactly how the machinery in their **Acquisition of the Data.**

charge is performing. The data required for constructing the curves are particularly easily got on war-ships, because the engines have to be so often indicated and the speeds vary so greatly. War-ships also travel as a rule at much lower speeds than merchant ships, so that it is generally not difficult to get some spots for the lower parts of the curve. However, these lower parts of the curve are of the greatest importance for the determination of the initial friction; the more lower spots are determined the nearer the initial friction found will approach the truth.

Practical construction of the curve.

13) For practical purposes at sea it is quite sufficiently accurate to take the revolutions per minute as abscissæ for the indicated thrust curve instead of the speeds of the ship, especially as these are not always known with the desirable exactness. Under ordinary circumstances the revolutions are a criterion of the ship's speed, for they are proportional to the speed of the screw, this being the product of the revolutions and pitch, therefore so long as the slip does not vary too much they are also proportional to the ship's speed. Of course the thrust curve constructed with the revolutions as abscissæ does not give absolute, but only relative values. In determining the initial friction the abscissa Oh is not to be divided in the proportion of 0.87 : 1 but to be bisected, because of the substitution of the revolutions for the ship's speed. Although such curves do not give the indicated thrust quite accurately they are of great practical use, because by their means the initial friction of the engine can be watched, particularly when a series of such curves for the engines under various conditions exists. If on comparing these curves, it appears that the ordinate OA is gradually increasing, this is a symptom of growing initial friction in the machinery. Some of the bearings may be getting out of truth, the balance-rings of the slides becoming leaky, or something of that kind. But other very important conclusions may be drawn from these curves as the following instance shews. A screw steamer on her trials exhibited a very considerable slip and the indicated thrust curve rose much too slowly for higher speeds, giving evidence that the thrust of the propeller did not increase nearly fast enough. From both circumstances, the great slip and the low indicated thrust at high speeds, it was immediately inferred that the propeller was deficient in surface, and it had accordingly to be exchanged for one of greater surface.

Curve of mean pressure.

14) It is shewn in 8) that the reduced mean indicated pressure p_{i_r} of a two or three-stage expansion engine is in a certain

exactly defined simple proportion to the indicated thrust *P*, so that the indicated thrust curve will also serve as a curve of this mean pressure by the use of a suitable scale. But the curve of indicated horse-power which is also proportional to the mean pressure (reduced) can also be used as a mean pressure curve by means of a second scale, as is shewn*) in Fig. 3 Pl. 10. Thus we may see at once from the diagram what reduced mean indicated pressure must be present to produce a certain *IHP* in the engine, *for one particular displacement of the ship.*

15) c. **The curves of slip**, i. e. which exhibit the difference between the speed of the screw and that of the ship, — *the apparent slip,* can be constructed by two different methods. The observed slips are placed as ordinates over the abscissæ of the corresponding speeds either in knots per hour or in percentages of the speeds of screw. In the first case the same scale is usually taken for the ordinates as for the abscissæ. In the other case some other scale is chosen, for instance as in Fig. 3 Pl. 10 where the unit of the ordinates is twice as great as that of the abscissæ. The curves of slip exhibit the efficiency of the screw at the various speeds. If these curves have sudden humps it is to be inferred that the screw is not suitable to the ship, and that either the diameter or the surface is wrong or even both. **Curve of slip.**

16) d. **The curve of revolutions** is constructed similarly to that of the *IHP*. The revolutions are drawn in to a certain scale as ordinates to the speeds as abscissæ. If the slip is constant, the ends of these ordinates lie in a straight line, but if not they form a curve of greater or less curvature according to the form of the slip curve, as is easily seen on reference to Figs. 3, 5, and 6, Pl. 10. The revolution curves giving the number of revolutions for every speed are of particular interest to officers of the watch on board war-ships cruising in a squadron, as they afford a guide for altering the ship's speed to a certain amount at any moment. **Curve of revolutions.**

17) e. **The curve of cut-off** is constructed like the others but with the cut-offs as ordinates, either in percentages or some other fractions of the stroke. For compound and multiple-expansion engines it is only necessary to construct the curve for H. P. cut-off as this is the only one that affects the steam consumption. The cut-off curve, especially with the ordinates measured in percentages, enables the engineer, by help of the convenient **Curve of cut-off.**

*) Speed curves. Engineering II, 1879. p. 344

valve-gear fitted to all recent marine engines, promptly to carry out an order from the bridge for any particular number of revolutions.

Curve of coal-consumption.

18) f. **The curve of coal-consumption** is obtained by putting in the coal-consumptions per hour or per knot at the different speeds as ordinates to some convenient scale in the diagram (see Fig. 6, Pl. 10). From these curves, based upon the *IHP* and speed, the coal-consumption can be predicted much more closely than by the ordinary rough method of estimating it (see 36. 8 to 10). The construction of such curves of the results of carefully observed trials cannot therefore be too warmly recommended to all responsible engineers.

Curve of performance-coefficients.

19) g. **The curve of performance-coefficients** (see § 37. 8) is constructed like the others, the ordinates being the values, corresponding to the differents speeds, of $\frac{v^3}{IHP}$ to a convenient scale. From this curve the performance-coefficients can be easily taken out, as they are products of $\frac{v^3}{IHP}$ and some constant. According to which constant is used, we get

$$\text{for the midship section coefficient } C = \frac{v^3 \mathfrak{A}}{IHP}$$

$$\text{„ „ displacement „ } C_1 = \frac{v^3 T^{2/3}}{IHP}$$

$$\text{„ „ augmented surface „ } C_2 = \frac{v^3 S_1}{IHP}.$$

By drawing in three scales (parallel to the y axis) the units of which are to the unit of the scale of the ordinates of the coefficient curve and to each other as $1 : \mathfrak{A} : T^{2/3} : S_1$, the magnitude of either of these coefficients can be read off the one curve, as shewn in Figs. 3 and 5, Pl. 10. If the work due to the ship's resistance varied exactly as the speed cubed, the curve of coefficients would be a straight line, so that its deviations from the straight shew where the above law of the variation of the resistance does not hold good. With the help of a curve of coefficients and that of the *IHP* it is no longer so difficult to determine the engine power for proposed new steamers. If a ship is to be built, whose form and dimensions agree in general with those of an existing one whose trial trip results are known, and if the speed of the proposed vessel is to be lower, the necessary engine power can easily be taken from the resistance-curves of the existing ship, after a small correction for the difference of initial friction. In the same way the power for a proposed ship of greater speed than an

existing one can be arrived at unless the new vessel is to have a very much higher speed, for which case the curves of the other will not be of any service.

20) The great advantages of progressive trials and the graphic representation of their results by means of curves as shewn in Figs. 3, 5 and 6, Pl. 10, may be briefly recapitulated as follows. **Advantages of the graphic method.**

1. It is possible to see at once the required horse-power, the mean pressure reduced to L. P. piston, necessary to produce it, the revolutions, the cut-off, the coal-consumption, and the slip that may be expected for any speed of ship (under the same conditions as at the trial-trip).
2. The determination of the initial friction shews the efficiency of the machinery at low speeds and enables it to be estimated for higher speeds as well as to be compared with that of other engines.
3. The efficiency of form of the ship may be inferred from the curves of indicated thrust and *IHP*. If these rise suddenly at higher speeds, it is evidence that the resistance is disproportionately increased and that the form of the ship is unsuitable for speeds so great.
4. For similar ships the horse-power can be determined from the curves of their predecessors (see 19).
5. The curves of slip under certain circumstances assist the detection of an unsuitable propeller (see 15).

Seventh Division.

Selection of the most Suitable Type of Engine.

§ 41.

Classification of the Types of Engines.

Classification of marine engines.

1) Marine engines may be classified
   a) According to the kind of propeller they actuate,
   b) According to the pressure of and manner of using their steam,
   c) According to the number, position, and attachment of their cylinders,
   d) According to the method of transmitting the force from the piston to the crank.

Class a) conditions the *position* of the machinery, class b) the *system*, and c) and d) the *type* of it.

Classification according to propeller.

2) a. **According to the kind of propeller** marine engines are divided into *screw, paddle, and reaction or "hydraulic" engines.*

Systems of marine engines.

3) b. **Classified according to pressure and manner of using the steam** we may distinguish

Low-pressure engines when the working pressure is below 2 atmos.,
Medium pressure engines „ „ „ „ „ between 2 and 4 atmos.,
High pressure engines „ „ „ „ „ above 4 atmos.,
Non-condensing engines, when the exhaust steam escapes to the atmosphere,
Condensing engines „ „ „ „ is condensed,

Single-expansion engines when the steam expands in one cylinder,

Double-expansion or compound engines, when the steam expands by successive stages in two cylinders,

Triple-expansion engines when the steam expands by successive stages in three cylinders,

Quadruple-expansion engines when the steam expands by successive stages in four cylinders. More than four stages have not hitherto been applied to marine engines.

4) Double-expansion engines are further subdivided into Subdivisions of double-expansion engines.

Double-expansion engines without receiver or Woolf's engines, whose pistons reach their dead-points simultaneously so that the steam passes directly from the first into the second cylinder,

Double-expansion engines with receiver, or compound engines whose pistons do not have simultaneous dead-points, so that the steam cannot pass directly from the first into the second cylinder but must remain during part of the stroke in a space between the two, called the receiver.

5) The former of the above subdivisions may be again separated into Subdivision of Woolf's engines.

Woolf's engines with similarly moving pistons, in which both pistons either drive one crank or two separate cranks with no angle between them, and

Woolf's engines with contrary moving pistons, in which the cranks are 180° apart.

6) Compound engines may be distinguished as Subdivision of Compound engines.

Two-cylinder compounds, with one high and one low-pressure cylinder, and

Three (or more) cylinder compounds, with one high and two or more low-pressure cylinders in which the steam, on passing from the high-pressure cylinder, expands simultaneously.

7) c. **According to the number, position, and attachment of the cylinders** the following types of engine exist. Types of engines classified according to the cylinders.

One, two, and three cylinder engines, and those with more cylinders, as well as,

Horizontal engines,

Vertical engines,

Diagonal engines, the cylinders being inclined,

Steam hammer or inverted direct-acting engines when the cylinders are placed vertically over the crankshaft,

Fixed engines, when the cylinders are fixed,

Oscillating engines, when the cylinders move on trunnions.

8) d. **According to the method of transmitting the force from the piston to the crank**, the following types may be specified, Types of marine engines according to the method of transmission.

Direct-acting engines, when the piston rod is attached to the connecting rod by a cross-head,

Indirect acting or beam (or side lever) engines, when the piston-rod acts upon the connecting rod by a beam or lever.

9) The former type may be again divided into Subdivision of direct-acting engines.

Engines with solid piston-rods, when the connection between piston-rod and connecting rod is outside the cylinder,

Engines with hollow piston-rods or *trunk engines*, when this connection is made inside the cylinder;

or into

Engines with direct acting connecting rod when the connecting rod is placed between the cylinder and the crank-shaft,

Engines with return-connecting rod (steeple-engines) when the crankshaft is placed between the cylinder and the connecting rod.

§ 42.

Characteristics of the best Type of Engine.

Best system of machinery.

1) The best system of machinery is that one which produces, with
 a) the greatest economy,
 b) the greatest useful effect and
 c) the greatest uniformity of working.

Economy of marine-engines.

2) a. **The economy of a marine engine** is measured (as explained in § 35, 6) by the steam consumption per *IHP* per hour. As the same power can be obtained with high initial pressure and early cut-off as with low initial pressure and late cut-off and as the specific weight of steam rises in a lower ratio than as the pressure (see Steam-table p. 28 et seq.), it follows that the steam consumption per *IHP* per hour will be smaller, the higher the pressure and the earlier the cut-off. Other advantages of high-pressure steam are detailed further on in § 44. But, other things being equal, the hourly steam consumption per *IHP* depends upon the losses of steam in the engine (see Division III). From the economical point of view therefore, marine engines of high working pressure, early cut-off, and small steam-losses are the best.

Useful effect of marine-engines.

3) b. **The useful effect of a marine engine** is expressed by the quotient

$$\eta = \frac{SHP}{IHP},$$

the numerical value of which for screw engines of the present day (see § 30, 23) varies between 0.45 and 0.5, rising in rare cases perhaps as high as 0.55. The greater this quotient comes out, the more of the *IHP* of the engines is devoted to driving the ship, and the greater therefore is her speed, other things being equal. If we disregard the work due to the propeller resistance, which *must* be done in comparing different systems of engines, and consider as the effective performance of the engine, the work transmitted by the crank-shaft, this latter will be the greater, the smaller the work lost in friction is, compared with the indicated work performed by the steam in the cylinder. Thence follows that, judging engines by their useful effect, that engine is the best one in which the ratio

$$\frac{\textit{Friction work}}{\textit{Indicated work}} \text{ is the smallest.}$$

4). c. **The uniformity of working of a marine engine** depends upon the following circumstances. The tangential load $\frac{P\, sin\,(\alpha + \beta)}{cos\, \alpha}$ (see further on in § 78, 3) acting in the crank-pin circle of the engine has in every revolution two minima, corresponding to the dead-points in which it is zero, and two maxima at the points where the connecting-rod is tangential to the crank pin circle, where this load becomes equal to the load along the connecting-rod $\frac{P}{cos\, \alpha}$. Between these limits the tangential load imparts an accelerated or retarded motion to the crank, according to its position, so that the peripheral velocity of the crank-pin circle varies during one revolution between V_{max} and V_{min}. Uniformity of Working.

5) If the average peripheral velocity of the crank-pin circle is Degree of uniformity.

$$\frac{V_{max} + V_{min}}{2} = V \text{ m. per sec.}$$

then the quotient

$$i = \frac{V}{V_{max} - V_{min}} \quad \ldots\ldots\ldots\ldots\ldots \quad (212)$$

which is called *the degree of uniformity* affords a measure of the more or less uniform working of the engine. The degree of uniformity will be the greater the smaller the difference between V_{max} and V_{min}, or the difference between the greatest and least tangential loads.

6) At high piston-speeds, high pressures, and early cut-offs, the difference of tangential loads is influenced by the reciprocating motion of the pistons, piston-rods, connecting rods, &c., or in short by the weight of these rods (see further on § 48). According to RADINGER*) this weight for recent triple-expansion engines in war-ships, due to piston and rod, cross-head, connecting rod, and half of the crank for every cylinder combined and referred to L.P. piston, amounts to about 3 kg. per sq. cm. of its area. For the heavier engines of merchant ships it will somewhat exceed this figure, while for torpedo-boat engines it is as low as 1.5 kg. per sq. cm. of L.P. piston area. This weight of rods must be regarded in determining the tangential loads. As an instance of the influence of the rods, ZIESE quotes the case of the three-cylinder compound engines of the British Iron-clad "Dreadnought" in which, when they were developing their full power of 8000 *IHP* with a piston speed of about 3 m. per sec., the stored-up energy in the pistons and rods for every half stroke corresponded to Influence of the rods.

*) J. J. RADINGER. Ueber Dampfmaschinen mit hoher Kolbengeschwindigkeit. Edition. III. Vienna 1892. Table III.

about 400 *IHP*. In the two cylinder compound engine of a torpedo-boat, indicating 500 to 600 horse-power at 4.5 m. per sec. piston speed, 73 horse-power was absorbed in the same manner.

Variations of load.

7) The variations of load on the crank transmitted to the screw shaft by this important influence of the weight of the rods, unless equalized by some special arrangement (particularly with high working pressure and high ratio of expansion), probably affect the material of the shafting as if it were exposed to a number of constantly recurring small blows, that is they gradually render it crystalline. This circumstance perhaps explains the frequent cases of fractured shafts on large screw steamers of late years. In general it may therefore be said that the more uniform the working of an engine is, the smaller will be the fatigue upon its various parts, the less the wear and tear, and the longer it will run without requiring serious repairs, a matter of inestimable importance for long voyage steamers.

Most profitable system.

8) From the foregoing it may be inferred that the most profitable system is that under which

a) the steam consumption per *IHP* per hour,

b) the ratio of the frictional work to the indicated,

c) the variation in the tangential loads

are the smallest. From these points of view, single, double, triple, and quadruple expansion engines will be subjected to investigation in the succeeding paragraphs.

§ 43.

Advantages of Surface-condensation and High-pressure Steam.

Watt's Low-pressure engines.

1) **I. Early steamers.** In the middle of the fifties by far the greater number of the existing screw steamers were fitted with WATT'S low-pressure engines with a working pressure of 1 to 2 atmospheres. Their cylinders were not jacketed and the usual reversing gear was STEPHENSON'S link-motion, only employed to vary the cut-off within very modest limits. But few engines could boast of a clumsy expansion-valve, generally arranged so that it could be thrown out of gear. The jet-condenser was worked with sea-injection. The engines were often geared, only the more recent ones had PENN'S direct arrangement.

Speed and consumption of early screw-steamers.

2) The faster ships fitted with the latest and best engines of the above description attained on an average a trial-trip speed of eleven to twelve knots. Fourteen knots was then considered a very respectable speed for screw-steamers and was only reached on the measured mile, whereas at the end of the

forties (for instance the "Banshee" in 1847) nearly 16 knots was already obtained on measured mile runs with paddle boats. The consumption at that time was about 2.5 kg. per *IHP* per hour and in exceptional cases with large engines and skilful firing might be got down as low as 2.2 on trial trips. So that 30 years ago a steamer indicating 1000 horse-power on an average, required for only a 20 days run a stock of 1200 tons of coals and adding 10 to 20 % for contingencies, say 100 to 200 tons more. The total bunkers thus amounted to from 1300 to 1400 tons, which, considering the tonnage of the steamers of that day, was a very heavy drain on the ship's carrying capacity and shewed little prospect of successful competition with sailing ships to ports where no coal was to be obtained.

3) The further extension of steam navigation depended therefore to such an extent on bringing down the consumption that engineers in the fifties were forced to try every improvement in boilers or engines which shewed a chance of even the smallest saving of coal. Among these improvements were superheating the steam (see § 13, 13) and the re-introduction of the jacket (see § 16, 11). But not till during the sixties were those incisive and revolutionary changes introduced which were followed by really important reductions of the coal-consumption. In the first rank of these was the invention of the surface condenser and the application of high-pressure steam to marine engines thus rendered possible. Soon afterwards the compound engine, later still the triple-expansion engine, and quite recently the quadruple-expansion engine were tried with the object of more completely reaping the benefit of high-pressure steam. As the following investigations will shew the problem throughout is only the more favourable employment or rather the more thorough utilizing of the steam after it has once been generated, and in this direction brilliant progress has undoubtedly been recorded. **Progress in the application of steam.**

4) **II. Advantages of surface-condensation.** The advantages of the surface-condenser, compared with WATT'S jet-condenser, may be ranked as follows, if taken in the inverse order of their importance: **Advantages of the surface-condenser.**

a) they produce a better vacuum,
b) they give purer feed-water,
c) they enable high-pressure steam to be carried.

5) **a. Better vacuum with surface-condensers.** Whereas in ordinary working the vacuum gauge of a jet-condenser shews 0.8 to 0.85 kg. per sq. cm., that of a surface condenser stands as a rule at **Better vacuum.**

0.9 to 0.95 and above. The lower vacuum in the jet-condenser is due to the air carried into the condenser with the injection water. In the surface-condenser the steam does not come into contact with the cooling water, and apart from leaks, the only outside source of air in the condenser is the supplementary feed when this is taken from the circulating water into the condenser in order to heat the water (see § 33, 7).

Objections to salt feed-water.

6) b. **Purer feed-water of the surface-condenser.** Seawater contains between 3.3 and 3.5 % of solids in solution which remain in the boiler after evaporation (see 33, 3). Things are not much improved by taking the feed from a jet-condenser, as the steam requires 20 to 30 times its weight of injection water according to the temperature of this latter (i. e. the latitude) to condense it. In the most favourable case therefore, 1 kg. of fresh water arising from the condensed steam is distributed over 20 kg. of injection water, so that the feed-water from a jet-condenser still contains over 3 % of matter in solution. Among the constituents of this, carbonate and sulphate of lime, of which sea-water contains nearly 0.2 %, are the most dangerous, because they become insoluble at so low a temperature as 144° C. They are completely precipitated at this temperature without reference to the degree of saturation of the solution and cover the internal parts of the boiler with the well-known scale. Boiler scale being such a bad conductor of heat, a uniform thickness of 2 mm. on all the heating surface is sufficient to increase the consumption 20 % above what it is with clean surfaces; and this thickness would be reached in boilers of 200 sq. m. heating surface, fed from a jet-condenser, within 7 hours. But the risk is more serious when the scale attains considerable thickness and, by removing the furnace and combustion chamber plates from the cooling influence of the water, causes them to become heated. As, when hot, the plate has a considerably lower strength than when cold, the pressure in the boiler forces in the furnace and combustion chamber crowns, a highly dangerous occurrence.

Waste of Fuel by feeding from a jet-condenser.

7) For the above reasons the temperature of the boiler water with jet-condensers had to be kept below 144° C. to prevent the deposition of salt. As a rule 135° C. were not exceeded, corresponding to a pressure of 2 atmospheres. This precaution alone was not sufficient however; the percentage of salts held in solution in the boiler water was obliged to be kept below a certain limit, because the salt contained in sea-water crystalizes out in large quantities at 135° C., when present at a strength of 12 %. The saltness with jet-condensing engines is now kept

at about 9 % in order to prolong the life of the boilers as much as possible, so that half the weight of water evaporated has to be blown off by Eq. 151, p. 284, when the feed-water contains 3 % of salt. As shewn in § 35, 4 the waste of fuel then amounts to from 6 to 7 %, which is a very favourable state of things for jet-condensers. In practice the waste rises to about 15 %, with low temperature of feed, higher saltness of the sea, and an ordinary quantity of scale in the boilers.

8) When feeding from a surface-condenser we have to do in the first instance with pure distilled water; but as it is not sufficient to keep up the water level in the boiler on account of the various leaks, a certain quantity of other water must be added to it. As already mentioned this supplementary feed-water was usually taken from the circulating discharge at the condenser. In well kept engines the supplementary feed is about 2 % of the water condensed, see § 33, 8, but even taking a very high estimate, say 10 %, the saltness of the feed-water including the supplementary feed of 3.3 to 3.5 % saltness, would at the most be about $^1/_3$ %. In such feed-water there are only traces of objectionable lime-salts, so that blowing can be almost entirely given up, the more so as at moderate working pressures the saltness of the boiler water is allowed to rise as high as 12 % in order to get a thin scale on the boilers to protect them against corrosion. As less blowing off became necessary the coal-consumption of the marine engine fell also. On trial-trips of low-pressure engines fitted with surface-condensers, jackets, and superheaters it was about 1.5 to 1.6 kg. per *IHP* per hour, thus shewing a saving of 15 to 20 % over similar engines with jet-condensers, consuming about 1.9 to 2 kg. per *IHP* per hour. But compared with the best of the old WATT engines the economy was fully 30 %. **Diminution of the waste of fuel with surface-condensers.**

9) **c. Possibility of high working pressure with surface-condensers.** Having regard to the very small percentage of dissolved matter in feed-water coming from surface-condensers, even when the supplementary feed is taken from the sea, the temperature of the boiler water can be raised to any desired degree without danger, which is the chief advantage of the surface condenser as *it renders possible the use of high-pressure steam at sea.* **Increase of temperature of boiler water.**

10) But in order to keep out of the boilers even those slight traces of dissolved salts taken into the feed-water by the use of salt supplementary feed, it has become usual of late years to fill the boilers with fresh water, to change it as rarely as possible, and sometimes to carry a store of fresh water in the watertight bottom compartments of war-ships and the ballast **Feeding with distilled water.**

tanks of traders (see § 33, 9). But with steam temperatures of 180 to 200° C. as in the latest marine boilers this precaution seems to be of little use as it does not prevent the formation of scale. It is a fact that since the introduction of triple-expansion engines, that is since high pressures in marine boilers have come into more general use, merchant steamers have frequently returned home with their furnace crowns down. Thanks to the good material now universally used in boilers, several cases of unmistakeable deformation of furnaces have occurred without tearing of the plate taking place, whereby not only was the approaching danger of explosion avoided, through prompt drawing of the fires, but the voyage could be completed under reduced pressure. Quite a thin layer of deposit has thus come to be regarded as capable of retarding the transmission of heat from the furnace gases to the water and thus heating the plates, after which their strength, already diminished by the high temperature of the steam, is no longer sufficient to withstand the working pressure. The boilers must therefore be kept absolutely free from internal deposit which can only be achieved by using distilled supplementary feed-water.

Advantages of high-pressure steam.

11) **III. Advantages of high-pressure steam.** The benefits expected to accrue from the application of high-pressure steam were referred to

a) equal coal consumed for equal weights of steam generated, whatever the pressure,

b) the specific weight of steam increasing at a slower rate than the pressure,

c) the greater expansive force of steam at higher pressure.

Reduced coal-consumption.

12) a. **Reduced coal-consumption with high-pressure steam.** When REGNAULT in 1847 had completed his investigations into the nature of saturated steam, the results of which are still as authoritative as ever, it was known that WATT'S old law of 650 T. U. being the constant heat quantity necessary to produce 1 kg. of steam of whatever pressure, is incorrect and that the total heat does increase with the pressure. Within the usual range of working pressures however, the total heat increases so slightly as scarcely to impair at all the practical utility of WATT'S law. According to REGNAULT 647 T. U. are required to convert 1 kg. of water at 0° C. into steam of 3 atmos. pressure; to take the pressure up to 6 atmos. 654.68 T. U. are required, i. e. 7.68 T. U. more. Out of the total quantity of heat produced by 1 kg. of average coal, about 5500 T. U. can be utilized in a well-designed

marine boiler, so that to convert 100 kg. of water at 0^0 C. into steam at 3 atmos. absolute pressure $\frac{647 \times 100}{5500} = 11.76$ kg. of coal are required, and to raise the the resulting steam to 6 atmos. absolute pressure $\frac{7.68 \times 100}{5500} = 0.14$ kg. more. This means an increase in the consumption of $1.19\,^0/_0$ or 1 ton in 84 to convert the same weight of water into steam of double the pressure. Such a small difference in the consumption is obviously imperceptible in practice. It is true that the higher the pressure, the greater this difference becomes, but it always remains so trifling that, for instance, steam of 12 atmos. absolute pressure requires only 1 ton in 40 more coal to produce a certain weight of it than steam of one-fourth the pressure.

13) b. **Comparatively light weight of high-pressure steam.** A cubic metre of steam at 3 atmos. abs. pressure weighs 1.603 kg., a cbm. of steam at 6 atmos. abs. pressure weighs 3.074 kg. If the weight of unit volume of steam increased proportionately with the pressure it would evidently be $1.603 \times 2 = 3.206$, but it is nearly $4^1/_2\,^0/_0$ lighter. The higher the pressure rises the more noticeable this difference becomes, for steam at 12 atmos. pressure is $8\,^0/_0$ lighter than it would be if its weight increased directly as the pressure. **Smaller weight.**

14) c. **Greater expansive force of high-pressure steam.** On calculating the theoretical work performed by steam of 3 atmos. absolute pressure during expansion in the cylinder of an engine, the cut-off being at $35\,^0/_0$ of the stroke, as shewn by experience to be most economical, it will be found that the same amount of work can be got out of steam of 6 atmos. absolute pressure, cutting off at $11\,^0/_0$, and steam of 12 atmos. absolute pressure cutting off at a little less than $5\,^0/_0$, the other conditions being the same in all three cases. **Greater expansive force.**

§ 44.

Single-expansion Engines.

1) Single-expansion engines may be classed as **Classification.**

I. Low-pressure engines,
II. High-pressure engines;

and these again as

a) Single-cylinder engines,
b) Two-cylinder engines,
c) Three-cylinder engines.

Low-pressure engines.

2) **Low-pressure engines.** The first marine engines built at the beginning of this century were single-cylinder low-pressure engines.

Single-cylinder, single-expansion engines.

3) a. **Single-cylinder engines.** In all single-cylinder engines the ratio $\frac{\text{Friction-work}}{\text{Indicated work}}$ is very large, while the crank is near the dead points, for the travel of the piston is here very small in proportion to the travel of the crank-pin and the load transmitted at this moment (during admission) by the piston which is a maximum, is resolved into a very small tangential and a very great radial load, as shewn in § 78, 3. Later on when the piston travel becomes greater in proportion to the crank-pin travel, the pressure in the cylinder rapidly diminishes in consequence of the expansion, and the ratio $\frac{\text{Friction-work}}{\text{Indicated work}}$ does not vary much for the rest of the stroke. It remains the greater in general, the greater the initial pressure and the earlier the cut-off, i. e. the greater the range of the piston-load during the stroke.

Objections to the single-cylinder engine.

4) A further disadvantage of the single-cylinder engine is that it will not start when on the centre, so that it can only be made available for marine purposes either by the use of balance-weights or some other arrangement to keep it from ever stopping on the centre. Nevertheless, as RADINGER has shewn, it is possible to attain a high degree of uniformity in the single-cylinder engine, even with high initial pressure and early cut-off, by means of a high piston-speed and taking advantage of the influence of the rods. But on the whole the addition of a second engine gives a higher useful effect and a more promptly manœuvring engine.

Two-cylinder engines.

5) b. **Two-cylinder engines** with single-expansion were originated by the above considerations. Their cranks are 90^{0} apart, so that the one piston is at about half stroke when the other is on the centre and vice versâ. If for these engines the loads transmitted to the crank-pin during one stroke are severally compared with those of a single-crank engine, it will be found that the range of these loads is smaller in the two-crank engine. From this it follows that the useful effect of the two-crank engine is the greater. These engines also possess the handiness so indispensably necessary for a marine engine in a higher degree, as with high pressures, they will start without difficulty in any position in which they may happen to have stopped. With low pressures however, it has been necessary in certain cases to resort to auxiliary starting valves. Uniformity of

running cannot be attained in two-crank engines as it can in single-crank engines, by choosing a piston-speed adopted to their cut-off, so that under certain circumstances a single-crank engine may run more uniformly than a two-crank engine of the same cut-off and initial pressure.

6) The low-pressure two-crank engine of 2 atmos. pressure was almost universally the rule for merchant ships in the sixties. Steamers fitted with it generally had surface-condensers, jackets, and superheaters. Their economy can be seen by the following table which gives the average working results on all voyages of most of the steamers of the North German Lloyd Co. built by CAIRD & Co. of Greenock between 1861 and 1869. **Coal-consumption of the two-crank engine.**

Ship's Name	Average IHP	Coal-consumption		
		daily in tons engl.	hourly per IHP in lbs. engl.	hourly per IHP in kg.
Hansa	1435.5	55.094	3.538	1.605
Weser	1734.9	68.064	3.662	1.661
Rhein	2192.9	71.961	3.063	1.389
Main	2056.4	74.273	3.371	1.529
Donau	1979.5	69.079	3.257	1.477
Mosel	2235.5	72.871	3.042	1.380
Baltimore	811.4	33.705	3.877	1.759
Berlin	1036.2	34.357	3.095	1.404
Ohio	1126.7	41.994	3.479	1.578
Leipzig	1147.7	42.500	3.456	1.567
Frankfurt	821.1	36.758	4.225	1.916
Hannover	928.8	36.322	3.650	1.656
		Average	3.476	1.577

The average coal-consumption is therefore in round numbers 1.6 kg. per *IHP* per hour, for the smaller engines it rises to nearly 2 kg. and for the larger ones it falls to about 1.4.

7) c. **Three-crank engines.** The difference between the greatest and least loads upon the crank-pins is still more reduced by using, instead of two cranks, three spaced 120° apart. Such three crank single-expansion engines were first fitted in the French Navy and later adopted by MAUDSLAY in the British Navy. They were also used in many German war-ships built at the end of the sixties and beginning of the seventies. The useful effect and degree of uniformity are greater in these than in two-crank engines, their handiness in manœuvring is also more marked, as they will start immediately from any position without the assistance of starting valves. **Three-cylinder engines.**

Advantages of three-cylinder engines.

8) Another very important advantage for large engines of several thousand horse-power consists in the reduction of the dimensions of the cylinders, if the same horse-power is to be got out of three cylinders instead of two under the same conditions in other respects. The casting, machining, and erection of very large cylinders are themselves attended with difficulties, which are increased when an old defective cylinder is to be taken ont of the ship and renewed. As besides, the three-crank engine is in general no heavier than the two-crank engine, because the weight of the third engine is made up for by the reduced dimensions of the other two, these engines are the best of single-expansion engines. Nevertheless, their comparatively high cost confined them chiefly to war-ships.

High-pressure engines.

9) **II. High-pressure engines.** Nearly simultaneously with the introduction of the surface-condenser and the recognition of its chief advantage of enabling high-pressure steam to be used at sea, high-pressure engines began to be built. At first, at the beginning of the sixties they were designed as single-expansion engines, mostly with two cranks, exactly upon the model of the hitherto existing low-pressure engines with surface-condensers, jackets, and superheaters acting as steam driers. They were but short-lived however, and never got into extensive use, because the advantages hoped for from the rise of working pressure from 2 to 4, then to 5, and finally to 6 atmospheres, were only partially realized, or at any rate remained a long way behind what was expected of them. This explains why the low-pressure engine with surface condenser, jackets, and superheater kept the field until the end of the sixties and was not displaced until the compound engine had unmistakeably established its superiority.

Coal-consumption of high-pressure engines.

10) For the theoretical quantities of work of high-pressure steam referred to in § 43, 14, the weight of feed at 6 atmos. abs. pressure and 11 % cut-off is about 40 % (43, 13), that for 12 atmos. abs. pressure and 5 % cut-off, more than 50 % less than that required at 3 atmos. abs. pressure and 35 % cut-off. Therefore the coal-consumption must be 40 % lower in the first and 50 % lower in the second case than in the last. Single-expansion high-pressure engines with working pressures of 4 to 6 atmos. however consumed *on trial-trip* 1.3 to 1.4 kg. per *IHP* per hour, so that they were only 12 % better than low-pressure engines of equal power. *In average work at sea* this economy came out even lower, as the following short statement of CAIRD'S North German Lloyd boats built in 1870/71 shews.

Ship's Name	Average IHP	Coal-consumption		
		daily in tons engl.	hourly per IHP in lbs. engl.	hourly per IHP in kg.
"Kronprinz Friedrich Wilhelm"	846.0	32.582	3.591	1.629
"Graf Bismarck"	875.0	34.112	3.638	1.650
"Köln"	955.2	36.623	3.578	1.623
		Average	3.602	1.634

On comparing with these the results of the low-pressure engines in the table on p. 381, we find for

"Hannover" at 928.8 *IHP* a consumption of 1.656 kg. per *IHP* per hour,
"Baltimore" „ 811.4 „ „ „ „ 1.759 „ „ „ „ „
"Frankfurt" „ 821.1 „ „ „ „ 1.916 „ „ „ „ „

Average 1.777 kg. per *IHP* per hour.

Thus the consumption of CAIRD'S high-pressure engines was only about 8% lower than that of their low-pressure engines of equal power.

11) The reasons which caused such a poor result were **Reasons of the low results of single-expansion high-pressure engines.**
   a) the smaller efficiency of cylindrical boilers,
   b) the abandonment of superheating,
   c) the great range of temperature in the cylinders,
   d) the increased loss of steam from leaks.

12) a. **Smaller efficiency of cylindrical boilers.** The box-boiler was impracticable at a greater working pressure than 2 atmos. on account of the difficulty of staying its flat sides. Cylindrical boilers were therefore introduced as possessing greater resistance, and for the same reason they were fitted with cylindrical furnaces if, as was most frequently the case, they had return-tubes. In the narrow furnaces of these boilers a less perfect combustion takes place than in the roomy ones of the box-boilers, so that the latter were the more rapid and economical steam-generators. Whereas in the box-boilers 8.5 to 8.7 kg. of water were evaporated by 1 kg. of good coal, in the cylindrical boilers an evaporative ratio of only 8.1 to 8.3 was attained. Thus the introduction of the cylindrical boiler caused an increase in the consumption of from 5 to 6% and this was the more marked the smaller the diameter and the greater the length of the cylindrical furnaces. **Smaller efficiency of cylindrical boilers.**

13) b. **Abandonment of superheating.** Another, and perhaps for ordinary working, equally important increase in the consumption was due to giving up superheating the high-pressure steam for the reasons already stated in § 13, 22. **Abandonment of superheating.**

Internal condensation.

14) c. **Internal condensation.** But the greatest waste of fuel arose from the important heat-losses which must always take place during the expansion of steam at high pressure and therefore high temperature in an *unjacketed* cylinder, producing the extensive condensation during admission spoken of in § 15.

In a jacketed cylinder this drawback can be avoided to a certain extent depending upon the efficiency of the jacket, but as here also the cylinder wall still tends to take the mean temperature, the wall must be retained on an average at the temperature of the admission steam by a constant transmission of heat from the jacket steam, causing condensation of the latter, which although not nearly so serious as the condensation in an unjacketed cylinder, is nevertheless by no means inconsiderable. Engineer Emery of the U. S. Navy determined the steam losses in the cylinders of the "Gallatin's" condensing engine of 4 to 5 atmos. working pressure at 30% of the total steam used with jackets off, and 20% with jackets on (see Table on p. 86).

Steam-losses due to leaks.

15) d. **The Loss due to passing steam** rises with the initial pressure, as the difference of the pressures on opposite sides of the piston and therefore the velocity of the steam passing the open places is greater. Although the loss from passing steam is in good engines only small compared with that due to internal condensation (see § 31, 27), still it detracts to some extent from the economy obtained from high-pressure steam.

Curtailing the Steam-losses.

16) In order to reduce the steam-losses it was necessary that all exertions should be made in the direction of diminishing not only the difference of temperature between the admission and exhaust steam but also the difference between the pressures on the opposite sides of the piston. Single-expansion engines could not answer to these requirements, and as the waste of fuel caused by steam-losses in the cylinder was increased by that due to the less efficient boilers and the want of the supheater, it is no wonder that the theoretical saving of 40% shrank to something like 12% in reality.

§ 45.

Double-expansion Engines.

Classification.

1) **I. Woolf's Engines.** The first double-expansion engine was constructed in 1781 by HORNBLOWER*) as a single-acting pumping engine, but had little success as the working pressure was too low for two-stage expansion. The steam entered a small high-

*) N. M. DOUGALL. The relative merits of simple and compound engines as applied to ships of war. London 1875. p. 9.

pressure cylinder, and having there expanded to a certain degree, passed directly into a larger low-pressure cylinder. In 1804 WOOLF converted this engine into a double-action one and added a WATT'S jet-condenser. These engines were quickly adopted in Germany and France where they were called "WOOLF'S engines", while in England they have been included in the designation "compound engines"; they may be divided into

a) single crank WOOLF'S engines,
b) two-crank WOOLF'S engines.

2) a. **Single-crank Woolf's engines** have been built on the most divergent designs with one low-pressure cylinder and one high-pressure cylinder placed either *above*, *(tandem compound) behind*, *beside*, *or within it*, vertically, horizontally, or diagonally arranged for screws, as well as having oscillating cylinders placed either side by side or one inside the other for paddles. Various as the plans are, they all may be divided into two great groups with *pistons moving in the same direction* and *pistons moving in opposite directions*. In all of them the H.P. exhaust passes directly into the L.P. cylinder. **Single crank Woolf's engines.**

3) *The advantages of the single-crank Woolf's engine* over the single expansion engine are **Advantages of Woolf's engines.**

α) greater economy, i. e. less steam consumed per *IHP* per hour, due to the reduced steam-losses,

β) greater useful effect at the same cut-off, or conversely earlier cut-off and therefore economy of steam for the same useful effect,

γ) lower maximum stresses on the rods.

4) α. **Greater economy.** In WOOLF'S engine the difference between the pressures during one stroke is less than in the single-expansion engine, therefore the useful effect must be greater (§ 44, 3) than in the latter. As the steam no longer expands down to the condenser pressure in the H.P. cylinder, but only down to the initial pressure of the L.P. cylinder, where it subsequently expands to the condenser pressure, not only is the range of temperature in each cylinder reduced, but the difference between the pressures on opposite sides of the pistons also — as compared with the single-expansion engine, so that the steam losses must be smaller and therefore the steam consumption less, as is clearly shewn by the "Bache's" trials (see Table p. 87). **Smaller steam-losses in Woolf's engine.**

5) β. **Greater useful-effect at equal cut-off.** Because of the smaller difference between the pressures on the pistons and the resulting increase of useful effect of this engine compared with the single-expansion engine, an earlier cut-off can be used for the same power and the economy thus again heightened. **Greater economy with equal power.**

Lighter maximum load in Woolf's engine.

6) γ. **Lighter maximum stresses on the rods.** If the power developed in a WOOLF engine were to be produced in a single cylinder of the same dimensions as the L.P. cylinder of the WOOLF, the load transmitted to the rods during admission would, because of the large piston area, be much greater, than the combined loads transmitted by the small H.P. piston at the same initial pressure and the equal L.P. piston at a much lower initial pressure. Thence follows that the rods, as well as the connecting parts between the cylinder and the crankshaft (bedplate, columns, &c.) can be made lighter in a WOOLF engine than a single-expansion engine, which almost makes up for the weight of the extra cylinder.

Defects of the single-crank Woolf engine.

7) *The defects inherent in the single crank tandem compound engine* are
δ) its unhandiness,
ε) its want of uniformity of running when cutting off early.

Unhandiness.

8) δ. **The unhandiness** of single-crank tandem compounds is explained by both their pistons being on the centre together, so that they are no better starters than one-crank single-expansion engines. Although in North America and in England, especially by STEPHENSON & Co., Newcastle-on-Tyne*), single-crank tandem compounds have been fitted, to save space in the ship, they are not at all to be recommended, as such engines with only one crank require a skilful and practised engineer to handle them.

Want of uniformity of running.

9) ε. **Their want of uniformity of running** when linked up in the H.P. is caused by the low pressure of the steam in the L.P. becoming insufficient to accelerate its heavy rods up to the normal piston speed. With later cut-off in the H.P. and moderate piston speed WOOLF engines work as evenly as single-expansion engines, and if vertically arranged with two cranks (180^0 apart) and pistons moving in opposite directions, — more evenly than compounds (with cranks 90^0 apart) on account of the better balancing of the rods. Such WOOLF engines have recently been much applied to driving dynamos on board ship.

Two-crank Woolf engines.

10) b. **Two-crank Woolf engines.** It was sought to impart the handiness of double engines to WOOLF'S by fitting a pair of them side by side to a shaft with two cranks 90^0 apart. The H.P. cylinder was placed either *behind*, *above*, or *inside* the L.P., so that both pistons of each engine travelled in the same direction.

Tandem engines.

11) *Woolf engines with tandem cylinders.* Where want of space, especially on twin screw war-ships, precludes the application of ordinary compound engines, a pair of horizontal tandem

*) The engineer 1866. I. p. 187. Illustration of the single crank tandem compound engine of the steamer "Prometheus".

compounds has often been selected in place of two or three-cylinder compounds, as for instance in the British despatch boats "Iris" and "Mercury"*), in the German iron-clad "Oldenburg" and the German cruisers "Irene" and "Prinzess Wilhelm". In the "Oldenburg" the H.P. cylinders are in the wings behind the L.P. cylinders and in the other ships in front of them.

12) *Vertical tandems* are fitted in the French mail-steamers "Normandie", "Bourgogne", and "Gascogne"**) the first built at BARROW the other two by the Societé des FORGES et CHANTIERS de la Méditerranée at LA SEYNE. These engines have three pairs of tandem cylinders working three cranks and indicated on trial 8000 *IHP*, the working pressure being 8 atmos. At sea they only reach about 6000 *IHP* and do not much exceed 16 knots average between Havre and New-York. **Vertical tandems.**

13) BUTLER***) of Cardiff makes single-acting WOOLF engines of special design from about 15 to 70 *IHP* for launches. The cranks of the two engines are 90° apart and steam is only admitted on the down stroke whereby the maker claims that noiseless running is obtained. The L.P. cylinders are open at the bottom and arranged like those of the well-known WILLANS engine. **Butler's launch engines.**

14) *Two-crank oscillating annular Woolf engines*, with the L.P. cylinders surrounding the H.P. were fitted in the Danish mail-steamers built by BURMEISTER & WAIN of Copenhagen (see "Aegir" in the table on p. 307). The working pressure is $5^1/_4$ atmos., the *IHP* about 900, and the consumption about 0.9 to 1 kg. per *IHP* per hour. **Oscillating annular engines.**

15) *Coupled Woolf engines with cylinders placed side by side* may be seen in the new Hudson ferry steamers of the Hoboken FERRY C°., New York. The cranks of each pair of cylinders are 180° apart and the cranks of the second engine are set at 90° from those of the first (making 4 cranks). The H.P. cranks are the leading ones and the vibration is very slight indeed. The boats have two screws, one forward and one aft, both right-handed. This arrangement renders the boat very handy, as the forward screw has the effect of stopping her almost instantly. This is the only reason why the new screw ferry-boats are displacing the old paddle boats more and more, for the daily coal-consumption of the former, working at $8^3/_4$ atmos. is only 14% less than that of the beam engines of the paddle-boats, on account of the long stoppages (10 minutes after every trip). Another point is that the old engines with their **Cylinders side by side.**

*) Sir THOMAS BRASSEY. The British Navy Vol. I p. 495. London 1882.

**) Annales industrielles. 1885. II. p. 216.

***) The engineer. 1886. II. p. 58 with illustration.

low revolutions require only one engineer and one fireman, as the engineer oils the machinery during the stops. But the faster-running screw engines have to have a greaser besides, whose pay at the high American rate of wages, forms such an important item, that the cost of working on the two systems is the same.

Draw-backs of coupled Woolf engines.

16) Besides their comparatively heavy weight, these engines have the following more or less special draw-backs. In the *tandem* arrangement sometimes adopted for mail-steamers, the L.P. piston is only accessible after removing the H.P. cylinder, and in the *annular* arrangement for oscillators considerable steam losses take place at the external and internal packing-rings of the L.P. piston. For *horizontal tandems*, the most suitable design for engines of this type in war-ships, the available breadth in the ship is often too limited. For these reasons it was sought to modify the single WOOLF engine with cylinders placed side by side, in such a manner as to combine its economy with the handiness of the two-crank engine. The result of these endeavours was the "receiver-compound" engine which was introduced pretty early for marine purposes and is known, in Germany and France, where the name "WOOLF" engine is still in vogue, simply as the "compound engine".

History.

17) **II. The two-cylinder compound engine.** BRÜCKMANN'S*) researches have clearly established the fact that in 1828 and 29 the engines of the steamers "James Watt" and "Hercules" built by JOHN COCKERILL were altered into compounds by ROENTGEN of Fijenoord, and that they were followed by similar new engines. ERNEST WOOLF, who is stated by LEDIEU**) to have invented the compound engine, now turns out to have been only ROENTGEN'S agent in England, so that ROENTGEN is to be regarded as the inventor not only of the compound but of the multiple expansion engine generally, as the idea of it is clearly included in his patent specification. For what reasons the compound engine did not make its way at that time, can no longer be definitely traced, perhaps the marine boilers of the period were not equal to the continued strain of the high pressure. From the year 1854 RANDOLPH ELDER & Co. of Glasgow made particular efforts to get the compound engine adopted in the British Navy. Nevertheless nearly twenty years passed before the views of those in authority there were sufficiently enlightened to recognize its economical superiority.

*) Zeitschrift des Vereins Deutscher Ingenieure. 1892. p. 941.

**) A. LEDIEU. Les nouvelles machines marines Tome II. p. 20, Paris 1879.

18) The ordinary compound is a double-expansion engine with cylinders placed side by side, each working one of the two cranks which are 90° apart. When the H.P. engine is on the centre the L.P. engine is at about half stroke and steam can pass from the H.P. into the L.P. cylinder. The steam then expands in front of the H.P. piston and behind the L.P. piston, so that after half a stroke, when the L.P. engine is on the centre, the steam has reached its lowest pressure for exhausting into the condenser. If now the L.P. piston were to begin another stroke, there would only be steam of the condenser pressure remaining in front of the H.P. piston which could pass into the L.P. cylinder. But the same pressure would exist in front of the L.P. piston, because this end of the cylinder is in connection with the condenser and as the steam behind the L.P. piston must expand during its forward stroke, the pressure would at last become much lower than that in the condenser. On the L.P. piston reaching half stroke, the exhaust from the H.P. engine, then on the centre, would follow into the L.P., a large and almost empty space, where the steam would suddenly expand, and after the remaining half of the stroke, pass to the condenser. As such a process cannot be uniform and favourable, the H.P. exhaust in the compound engine is always shut off from the L.P. cylinder during about half the stroke. It is therefore necessary to have a space between the cylinders to receive that part of the H.P. exhaust which cannot pass directly to the L.P. engine after the latter has cut-off. This intermediate chamber, — the receiver — generally surrounded the H.P. cylinder concentrically in the early engines and in modern ones is usually formed by an enlargement of the pipe conducting the H.P. exhaust to the L.P. steam-chest. **Arrangement of the two-cylinder compound-engine.**

19) In the compound the range of the piston loads is still smaller than in the WOOLF engine, its useful effect and uniformity of running are therefore superior. But this only applies to single WOOLF engines and not to a pair of them coupled, with cranks at right-angles. A comparison of the tables on pages 86 and 92 proves that the steam consumption per *IHP* per hour is considerably smaller than in the single-expansion engine. **Comparison of two-cylinder compounds with Woolf engines.**

20) The older compound engines all had a working pressure of 60 ℔s. per sq. inch engl. = 4.22 atmos. and mostly a piston-speed of 2 m. per sec. With English engines of this type the following consumptions were got in German mail-steamers.*) **Working results of early compound engines.**

*) HAACK & BUSLEY. Die technische Entwickelung des Norddeutschen Lloyd u. s. w. Berlin 1893. p. 125.

Table of the Coal-consumption of older Compound Engines.

Ship's Name	Engineers	Year of starting	Average IHP	Coal-consumption daily in tons engl.	hourly per IHP in lbs. engl.	hourly per IHP in kg.	Average for all engines of one maker
1	2	3	4	5	6	7	8
Amerika	Day	1871	1882.20	47.216	2.3413	1.062	
Hermann	„	1872	1587.89	48.990	2.8804	1.306	
Deutschland . . .	„	1874	1848.89	50.384	2.5435	1.154	1.174
Neckar	Caird	1874	2088.11	55.562	2.4841	1.127	
Oder	„	1874	2071.74	60.767	2.7376	1.242	
Strassburg	„	1872	1067.07	35.804	2.9907	1.356	
Moltke	„	1873	1309.88	40.945	2.9174	1.318	
Roon	„	1874	1530.98	45.306	2.7620	1.253	
Werder	„	1874	1623.33	44.115	2.5364	1.150	1.241
Braunschweig . . .	Steele	1873	1524.47	43.901	2.6878	1.219	
Nürnberg	„	1873	1227.24	39.393	2.9960	1.359	1.289
Hohenzollern . . .	Earle	1873	1446.49	42.509	2.7428	1.244	
Hohenstaufen . . .	„	1874	1293.38	43.069	3.1079	1.410	1.327
					Average	1.246	

So that these older compounds consumed on an average 1.25 kg. of coal per horse per hour and thus were about 25% better than the best of the low-pressure engines which burnt about 1.6 kg. (see Table on p. 381).

Working results of later compounds.

21) In later years the working pressure of the compound engine was raised from 4 to 7 atmos. and simultaneously the piston

Table of the Coal-consumption of later Compound Engines.

Ship's Name	Engineers	Year built	Working pressure in atmos.	Average IHP	Coal-consumption daily in tons engl.	hourly per IHP in lbs. engl.	hourly per IHP in kg.	Average for the engines of the same or nearly the same pressure
1	2	3	4	5	6	7	8	9
Bavaria	Flensburg	1879	4.92	735	16.800	2.133	0.967	
Saxonia	Caird	1880	„	898	18.552	1.928	0.874	
Rhenania	Howden	„	„	860	18.518	2.005	0.909	
Borussia	Reiherstieg	1879	„	778	16.972	2.036	0.924	
Holsatia	„	1880	„	776	17.054	2.051	0.930	0.921 kg per IHP per hour
Thuringia	Reiherstieg	1880	5.27	863	18.000	1.946	0.883	
Polaria	Wallsend Sl.	1881	5.62	1200	25.773	2.004	0.909	
Australia	„	„	„	1007	20.608	1.910	0.866	
Polynesia	„	„	„	1173	23.877	1.899	0.861	
Bugia	Vulcan	1882	„	2700	54.100	1.870	0.848	
Slavonia	Hawthorn	1883	„	1138	31.318	2.568	1.165	
Rhaetia	Reiherstieg	„	„	2589	52.150	1.880	0.852	0.917 kg per IHP per hour
Bohemia	Inglis	1881	5.97	1860	36.000	1.860	0.844	
Hungaria	Reiherstieg	1884	6.25	1028	21.814	1.980	0.889	
California	Wallsend Sl	1883	6.33	1210	28.221	2.176	0.987	0.909 kg per IHP per hour
						Average	0.914	

speed from 2 to 2.5 m. per sec., whereupon the above consumptions of German mail-steamers were observed.*) As compared with the older compounds therefore, a further saving of 25 % was accomplished, which was to be ascribed less to the rise of pressure than to the increased piston-speed and the generally superior design of engine, for the higher pressure of 1.5 atmos. by itself only brought about a saving of barely 1.3 %. *The consumption of a compound was taken at 1 kg. per horse per hour average.*

22) The chief reason of the economical superiority of the later compounds over the older ones is the reduction of the dimensions of the low-pressure cylinder. In the original large ones losses both of effect and of steam took place, the former from the greater initial friction of the engine in consequence of the heavy rods of the L.P. engine, the latter from the considerable internal condensation due to carrying the expansion too far, as well as from the large clearances unavoidable in such great cylinders with ordinary valve gear. For these reasons it is advisable not to make the L.P. cylinder more than 2.3 to 2.5 m. diameter, using two smaller ones if necessary, to get the volume required. Besides the advantage of easier machining and erecting, a considerable improvement in economy followed the introduction of the two small L.P. cylinders as instanced by the following results of several well-known mail-steamers. In July 1885 the INMAN liner "City of Chester" and the Cunarder "Bothnia" left New York on the same day and arrived at Queenstown together. The steamers are of about equal dimensions and the following table gives the particulars of the engines, shewing that the "Bothnia" with the considerably smaller L.P. cylinder was about 50 % more economical at the same power than the "City of Chester" with her huge L.P. cylinder and exaggerated expansion. On comparing the North German Lloyd steamer "Ems" with the "City of Chester" it is found that the former

Improvements in the later compound.

Table shewing the saving of fuel due to adopting small L.P. cylinders.

Name	Working pressure atmos.	Diameter H.P. m	Diameter L.P. m	Stroke m	Expansions	Average IP	Average speed in knots	Average coal-consumption for 24 hours tons
1	2	3	4	5	6	7	8	9
City of Chester	5.50	1.73	3.04	1.67	10	4600	13.5	108
Bothnia	4.00	1.52	2.64	1.37	5	3000	13.5	72
Ems	6.66	1.57	2 of 2.23	1.52	8	6000	16.4	130

*) HAACK & BUSLEY. Die technische Entwicklung des Norddeutschen Lloyd u. s. w. Berlin 1893. p. 133.

boat with two small L.P. cylinders, rather higher pressure, about 10 % more displacement, and 20 % greater average consumption, attained a speed more than 20 % higher, thus requiring less coal for the whole run from England to America than the "City of Chester". The wasteful compound engine of the latter has therefore been since replaced by a triple.

Economy of Woolf and compound engines.

23) The widely prevailing impression that the compound-engine is more economical than the WOOLF engine is a mistaken one. It is, on the other hand, a fact that many early stationary WOOLF engines, as quoted by O. H. MÜLLER jnr.*) work much more economically than many recent compounds. WIDMANN'S heat-trials of various marine engines, referred to on pp. 120 and 121 also corroborate this statement. In these experiments the heat carried off with the exhaust steam into the condenser varies between 0.5 and 12.1 % of the total heat supplied to a WOOLF engine, whereas for four compounds it is between 6.7 and 21.9 %. This difference is explained by the friction in the passages between the cylinders and the cooling in the receiver to which the steam is exposed. These two important sources of loss are more easily avoided in the WOOLF than the compound and sometimes do not occur at all in the former.

Drawback to the compound.

24) With all its advantages the compound has one fault, viz. that it does not handle so well as an engine with two independent cylinders. It will only start readily when it has happened to stop before the H.P. engine has cut-off, so that when the stop-valve is opened the piston gets the full pressure of the steam. In all other positions it is usual to admit steam from a branch off the main steampipe by means of a hand starting-valve either into the L.P. cylinder or into the H.P. cylinder and the receiver, in order to start the engine, and as a certain time is required to move this valve the engine must handle less promptly, as stated above.

Three-cylinder compounds.

25) **III. Three-cylinder compounds.** With the ever increasing dimensions and speeds of ships, the horse-power rose to such an extent that it was no longer possible to get it out of one H.P. and one L.P. cylinder, as the latter became unmanageably large (see 22). Two L.P. cylinders were therefore adopted instead of one, and thus were evolved the still existing three-cylinder compound engines of many large war and merchant steamers.

Arrangement of the three-cylinder compound.

26) These engines are usually designed with the H.P. cylinder in the middle, that is, with one of the L.P. cylinders on each side of it. The arrangements of the cranks vary considerably.

*) Zeitschrift des Vereins deutscher Ingenieure. 1886. p. 820.

The cranks are generally 120° apart, particularly when the engines have often to work as single-expansion engines, as in war-ships.

27) In the comparatively early engines of the British cruisers "Bacchante" and "Boadicea" of 5250 *IHP*. RENNIE placed the H.P. crank so that it formed an angle of 90° with each of the L.P. cranks (Fig. 2, Pl. II), in order as far as possible to minimize the stresses on the rods and obtain an uniform tangential load. These engines worked smoothly and evenly at their highest as well as at lower powers and RENNIE*) therefore claimed that his crank arrangement was the best hitherto adopted. To prove this he constructed diagrams of tangential loads for various arrangements of cranks (see Figs 1, 2, 3 and 5, Pl. II) as adapted to the conditions of the engines of the ships named above. These conditions are, initial pressure $p = 5.5$ atmos., back-pressure $a = 0.25$ atmos., H.P. cut-off $\varepsilon = 0.5$, L.P. cut-off $\varepsilon = 0.6$, length of connecting rod $l = 48$. From these diagrams (Fig. 7, Pl. II) we may infer that **Rennie's Investigations.**

a) in RENNIE'S crank arrangement (Fig. 2) the range of tangential load was the smallest,
b) the next in order was the usual arrangement at 120° apart (Fig. 1),
c) then the arrangement shewn in Fig. 5 which has been only rarely used,
d) the worst was the French arrangement, Fig. 3.

The tangential loads for crank-arm 0.61 m are as follows

Crank-arrangement	Maximum load	Minimum load	Mean load	$\frac{\text{Maximum load}}{\text{Mean load}}$
a	150 Tons	95 Tons	119 Tons	1.26
b	167 "	65 "	119 "	1.40
c	185 "	68 "	119 "	1.55
d	190 "	55 "	119 "	1.60

In spite of these data it is an open question whether RENNIE'S arrangement still remains superior to that with the cranks at equal angles, when the distribution of the steam is chosen more suitably to the latter, as the tangent loads must then become more uniform.

28) There are in the French Navy three-cylinder compounds having the two L.P. cranks 90° apart and the H.P. crank 135° from each of them (Fig. 3, Pl. II). The old German cruisers "Carola" and "Olga" of 2100 *IHP* have the same crank arrangement **Experiments in the German Navy.**

*) Transactions of the Institution of Naval Architects. 1814. p. 156.

which possesses the advantage of enabling the engine to be worked as an ordinary two-cranked one with the two L.P. cylinders, in case the H.P. is injured and has to be disconnected. On a six hours' trial for the purpose of comparing the two conditions, the "Olga", working compound at 5 atmos. boiler pressure indicated 2393.9 IP for 13.92 knots, and 1060.5 IP for 11.41 knots with the H.P. engine disconnected and 3 atmos. boiler pressure.

Crank-arrangement of Humphrys and Schichau.

29) The engines of the British iron-clad "Dreadnought" of 8000 IHP by HUMPHRYS, TENNANT & Co. have their H.P. crank at 90° from one L.P. crank and the second L.P. crank 135° from each of the other cranks (Fig. 4, Pl. II). The old German cruisers "Habicht" and "Möwe" of 600 IHP built by SCHICHAU of Elbing have a similar crank arrangement, as shewn in Fig. 6 Pl. II. This last plan accomplishes the utmost possible equality in the power of the three cylinders, so that each does one third, whereas on the French system the H.P. cylinder does considerably more work than either of the L.P. cylinders, unless they are fitted with special expansion slides.

Receivers of three-cylinder Compounds.

30) Three-cylinder compounds have, as a rule, no special receivers, the volume of the H.P. exhaust passages and L.P. steam chests being sufficient for this purpose.

Advantages of three-cylinder Compounds.

31) These engines have in general a greater useful effect and greater uniformity of working than two-cylinder engines and FRÉMINVILLE*) states that their losses of effect are smaller and the influence of the clearance spaces less than in two cylinder compounds.

Working Results of three-cylinder Compounds.

32) FRÉMINVILLE'S assertions are corroborated by the average results of three-cylinder compounds fitted in large German mail steamers given in the following table.**)

Table of the coal-consumption of three-cylinder Compounds.

Ship's name	Engineers	Built	Working pressure atmos.	Average IHP	Coal-consumption daily in tons engl.	hourly per IHP in tons engl.	hourly per IHP kg
1	2	3	4	5	6	7	8
Elbe	Elder, Glasgow	1881	5.27	5200	122	2.13	0.97
Hammonia	Thomson, Glasgow	1882	5.62	4125	87	1.94	0.88
Werra & Fulda	Elder, Glasgow	1883	6.33	6000	128	1.96	0.89
Eider & Ems	"	1884	6.68	6500	135	1.92	0.87
						Average	0.90

*) A. DE FRÉMINVILLE. Etude sur les machines compound, leur rendement économique et les conditions générales de leur fonctionnement. Ouvrage couronné par l'Institut, Académie des Sciences. Paris 1878. p. 54.

**) HAACK und BUSLEY. Die technische Entwickelung des Norddeutschen Lloyd u. s. w. Berlin 1893. p. 143.

Leaving out the Elbe, the oldest of these ships, on account of her low working pressure, we find the average coal-consumption of large three-cylinder compounds is 0.88 kg per horse per hour, or about 4% lower than that of the contemporary smaller compounds, by virtue of the increased working pressure and rise of piston speed from 2.5 to 3 m average (see 22), and 3% lower than two-cylinder compounds of the same working pressure as the large three-cylinder engines. The actual ratio of expansion which had risen to 8 in the last of the engines quoted in the table on p. 390, went back again to 5 or 6 in these three-cylinder engines (compare 22).

33) With the object of still further increasing the economy of the compound engine, the working pressure, which was at most 4 atmos. when the compound was first introduced, was gradually raised by the beginning of the eighties, to 9 atmos. But the improvement anticipated from this high pressure was not achieved as the range both of temperature and pressure in the cylinders was too great. The idea of expanding the steam successively in three cylinders instead of two, and thus utilizing it more thoroughly was then nearly reached.

§ 46.

Triple Expansion Engines.

1) **I. Introduction of the Triple-expansion Engine.** The merit of having constructed the first practicable triple-expansion engine belongs to the English engineer Dr. ALEXANDER KIRK. It was fitted in the S. S. "Aberdeen" by R. NAPIER & SONS of Glasgow and tried in February 1882. KIRK had already designed a similar and equally efficient engine in the summer of 1874 when engaged at JOHN ELDER & Co's. for the S. S. "Propontis". This engine worked at 8.8 atmos. pressure but turned out a failure because its water-tube boilers became totally useless after scarcely a year and a half's work, as the author has particularly described in another place.*) The same fate occurred to a triple-expansion engine designed by a Mr. FRANKLIN**) for the German S. S. "Sexta" built by W. GRAY of West Hartlepool and tried in September 1874; the boilers were here again unequal to the the high working pressure of 8 atmos. A third engine of this kind was made by DOUGLAS and GRANT for the First triple-expansion Engine.

*) C. BUSLEY. Die Entwickelung der Schiffsmaschine in den letzten Jahrzehnten. Berlin 1892. Edition III. p. 173.

**) Zeitschrift des Vereins deutscher Ingenieure 1886. p. 886.

steam yacht "Isa"*) but was not a success. The first triple engine completed after the "Aberdeen's" was that of the S. S. "Isle of Dursey" in 1883 built at the Wallsend Slipway by Mr. ALEXANDER TAYLOR, the designer of the "Isa's" engine, and the next was fitted by SCHICHAU of Elbing in the Bremen S. S. "Nierstein" at the beginning of 1884.

First German Triple.

2) The triple-expansion engine came into use in the German Navy in the summer of 1883 when Admiralitätsrath Görris experimentally converted the two-cylinder compound engine of the torpedo-boat "Scharf" belonging to the the now obsolete "Schütze" class into a triple. With this object the working pressure was raised from 9 to 10 atmos. and three new cylinders were fitted in place of the two original ones. After these alterations the *IHP* was 600 at 365 revs and 0.17 total ratio of cut-off, against 509 *IHP* at 355 revs and 0.25 total ratio of cut-off obtained on the forced-draught trials of the "Schütze". The latter's fully equipped maximum speed was 17 knots, "Scharf" attained 18 with her converted engine. "Schütze's" machinery weighed 23.36 tons or 43.9 kg per *IHP*, "Scharf's" 23 tons or 38.3 kg per *IHP*. The superiority of the triple over the compound, both in regard to its greater efficiency as a steam-user and to its lighter weight thus became so strikingly evident that German torpedo-boats were fitted with triples as early as 1883, whereas the compound was retained some years longer in all other navies.

Division.

3) According to their number of cylinders, triples may be classed as
   a) three-cylinder,
   b) four-cylinder,
   c) five-cylinder.

Three-cylinder Triples.

4) **II. In three-cylinder triples**, the steam on leaving the boiler does its work successively in one high, one intermediate, and one low-pressure cylinder, whence it passes to the condenser. The cylinders are usually placed side by side, the H.P. forward, the M.P. next, and the L.P. aft. This was the arrangement in the above-mentioned "Propontis", "Aberdeen", "Isle of Dursey", and "Nierstein". Some builders have since placed the H.P. in the middle still keeping the L.P. aft, which is more convenient for the exhaust. In either case each piston drives one of the three cranks set at 120° apart. In the yacht "Isa" and the torpedo-boat "Scharf" the H.P. and M.P. formed a tandem with its crank at 90° from the L.P. one. This arrangement has since been often adopted in converting compound engines to triples, as discussed further on.

*) Engineering. 1879. I. p. 195.

5) All of the more powerful steamers fitted with three-crank triples are subject to vibrations caused by the fore and aft oscillations of the centre of pressure of the forces due to the reciprocating motion of the moving weights of the engine. The reason of this is that the engine seat is relieved of weight at one end at that moment of the down-stroke when the load due to the acceleration of the motion of the rods (see § 48) equals the weight of the engine, because the load transmitted to the crank-pin from the piston is diminished by the above load, while the admission steam has its full effect upon the respective cylinder-cover. Simultaneously the engine seat receives an extra load at the other end due to the two pistons being on the up-stroke, so that an alternating fore and aft tripping couple is produced. **Vibrations of Steamers.**

6) KLEEN *) proposed a system based on extensive observation, by means of which a certain number of revolutions per minute could be calculated so as to differ as much as possible from the period of elastic oscillation of the ship, for the vibration is most marked when these synchronize. His system was worked out in the first instance for shallow-draught paddle tugs. The formula is **Kleen's system of reducing vibration.**

$$\frac{u}{u_0} = \sqrt{C E \left(\frac{H}{L}\right)^3 \frac{\delta}{W}} + \ldots \ldots \ldots \ldots (213)$$

where u is the number of elastic oscillations per minute of the ship,

u_0 the number of double oscillations of the ship per minute due to buoyancy,

C a constant = 16 for light draught paddle boats and measurements in metres,

E the modulus of elasticity of the material of the hull,

H the moulded depth of the ship,

L the length of the ship,

δ the breadth of a rectangle $\delta\,H$, the moment of inertia of which equals that of the $\mathfrak{M}$,

W the area of the load water line;

on the metric system we have

$$u_0 = 29{,}21 \sqrt{\frac{w}{d}} \times \frac{1}{\sqrt{T}} \ldots \ldots \ldots \ldots (213^a)$$

in which

w is the coefficient of fineness of the LWL,

d the displacement coefficient,

T the mean draught ex keel.

*) Zeitschrift des Vereins deutscher Ingenieure 1893. p. 1487.

C having been determined by observation for a certain class of steamer and u_0 calculated, we can find u which is the number of revolutions to be avoided.

Example.

7) A steamer has a length $L = 76.2$ m, beam $B = 10.89$ m, mean draught ex keel $T = 6.10$ m, depth of hold $H = 7.26$ m, area of $LWL = 605.7$ □m. Taking the modulus of elasticity $E = 17500000$ and the constant $C = 16$, $v = 0.73$, $d = 0.6$, and $\delta = 0.14$, then by Eq. 213ª, $u_0 = 23.951$ and by Eq. 213 $\frac{u}{u_0} = 7.548$ and $u = 180.8$. We see therefore that the engines and the propeller of this steamer should be so designed that the average working number of revolutions does not too closely approach 90.

Ziese's method for avoiding Vibration.

8) ZIESE*) proposes for the avoidance of vibration, to put the three cylinders as close together as possible, thus shortening the engine and bringing the three systems of rods more nearly into one plane, so that the tripping couple becomes less effective and therefore the tremour less marked. If then, according to ZIESE, the three cylinders are united into one strong body rigidly connected with the bed-plate, their mass is sufficient to withstand the effect of the shifting centre of pressure and the entire engine forms a stiff girder with substantial upper and lower members. When this strong and self-contained system is properly secured to the engine seating which must be well stiffened, it is not easy for any unbalanced forces to be set up so as to cause vibration, even at the highest speed. SCHICHAU of Elbing designs not only his celebrated torpedo-boat engines but also the machinery of large war-ships and mail-steamers on this principle.

Four-cylinder Triples.

9) **III. Four-cylinder Triples** with one H.P. cylinder forward, one M.P. cylinder aft, and two L.P. cylinders placed side by side between the other two, are recommended by SCHLICK**) with the object of reducing vibration. As in the case of a compound when the L.P. cylinder requires to be over 2.3 to 2.5 m in diameter, and is therefore replaced by two smaller ones, this arrangement is also advisable for a large triple, so that SCHLICK'S proposal adapts itself naturally to the circumstances. SCHLICK spaces the four cylinders at equal distances apart (Plate 2, Fig. 8) and has four cranks, those of the two L.P. engines being 90° apart for ease of handling, while the other two cranks are arranged at such angles that their pistons and rods, the weights of which

*) Zeitschrift des Vereins deutscher Ingenieure 1894. p. 140.

**) Transactions of the Institution of Naval Architects. London 1894. p. 359.

are calculated from those of the L.P's., act as balance weights to the latter and thus provide against the tripping couple.

10) Calling the H.P. and M.P. cylinders 1 and 2 respectively, the two L.P. cylinders 3′ and 3″, the weights of their pistons and rods P with corresponding indices, and assuming $P'_3 = P''_3$ the weights of the two L.P's. to be given, we can eliminate the effect of P''_3, the weight of piston and rods of the M.P. cylinder 2 by placing a crank exactly opposite to III″ and making the weight of its piston and rods $\frac{2}{3} P''_2$ (Pl. 2, Fig. 9). To balance the loads due to the inertia of P'_3, the weight for the M.P. cylinder 2 must be $\frac{1}{3} P'_3$ and the crank be placed opposite crank III′. Similarly the weight for the H.P. cylinder 1 must be $\frac{1}{3} P''_3$ and the crank be placed opposite III″ and $\frac{2}{3} P'_3$ on one opposite crank III′. The resultants of the two forces which belong to cylinders 1 and 2 respectively give the weights for each crank P_2 and P_1 respectively and the directions of these resultants enclose the angle at which the cranks must stand to each other. For very fast-running engines, whose slide-valves are comparatively heavy, the latter as well as the excentric rods &c. can be included in the balancing, whereby the small angle between cranks I and II is increased and all four cranks approach more closely to the symmetrical arrangement of dividing the circle equally. It is of course assumed that the cylinders are spaced at equal distances apart, centre to centre, but even with unequal spacing the method just explained is also applicable if P'_3 and P''_3 are divided in proportion to the spacing. The effect of horizontally moving masses can also be surmounted in this way which has a considerable advantage over the system of balancing with back-weights on the cranks, because these, although they are capable of eliminating the vertical effects of inertia, themselves set up equally objectionable horizontal ones. Those defects only which are due to the limited length of the connecting rods cannot be overcome without special means. Schlick's system of balancing.

11) Another arrangement of four-cylinder triple has often been adopted in engines converted from two-cylinder compounds and in replacing old compounds with new triples where considerations of space prevented lengthening the engine room. In the former case it has been an object to retain as many original parts of the engine as possible, so that only the cylinders and their accessories were renewed; in the other case it has been necessary Four-cylinder Tandem Triples.

to adhere to a two-cranked engine for the sake of accessibility of the moving parts. If the H.P. cylinder is placed over either of the other two the corresponding crank may have considerably more work to do than the other. To avoid this inequality, one of the three stages of the expansion was taken in two equal cylinders. In such engines the diameter of the L.P. cylinder did not generally exceed 2 m, so that the otherwise approved arrangement of dividing it into two was not advisable. The only alternative was to have two H.P. cylinders and place one of them over each of the others. Thus arose the four-cylinder triple with two cranks at right angles as fitted in the North German Lloyd Steamers "Hohenstaufen", "Hohenzollern" and "Habsburg" built by Earle in 1873 76 and the "Salier" by the Vulcan Co. of Stettin in 1891, their original engines being two-cylinder compounds.

Mail-steamers' engines.

12) **IV. Five-cylinder Triples.** When the North German Lloyd Co. ordered the S. S. "Lahn" of the FAIRFIELD Co. in 1886, it was a question of designing a triple-expansion engine, the L.P. cylinder of which, in order to secure the *IHP* determined on, viz 7500 average at sea, would have to exceed 3 m in diameter, with the accompanying objections in the way of inordinate weight of piston and rods &c. It was therefore obviously advisable to divide the one large L.P. cylinder into two smaller ones as in the three-cylinder compounds. But the difficulty was to arrange for three-stage expansion in four cylinders and still retain the three cranks for the sake of their smooth running. The desirableness of a symmetrical engine suggested the idea of splitting up the H.P. cylinder also into two halves, one of which was placed as a tandem over each L.P., the single M.P. cylinder working the middle crank. This was the origin of the five-cylinder arrangement for unusually large triples as fitted in the "Spree" and "Havel" besides the "Lahn" and afterwards by the FAIRFIELD Co. in the "Campania" and "Lucania" the largest twin-screw steamers. Each of the latter developed nearly as much power as the "Spree" and "Havel" together for they indicated about 25000 horse-power on trial, whereas the "Havel" and "Spree" attained nearly 12800 in one engine.

Advantages and Disadvantages of Five-cylinder Engines.

13) In all multiple-expansion engines the pressure in the L.P. cylinder limits the piston-speed inasmuch as it is impossible to reduce the weights of pistons and rods any further because they are now only $^1/_5$ and even in some cases as light as $^1/_{10}$ of the weights adopted in stationary engines ashore. If very high piston-speeds are desired in large engines, they can only be safely obtained by increasing the pressure absorbed by the inertia of the L.P. piston and rods which is most simply arrived

at by placing two H.P. cylinders over two L.P's. as is done in the five-cylinder triple. The H.P. steam thus comes to the assistance of the L.P. rods and in the M.P. cylinder the initial pressure is always great enough to be without any limiting influence upon the piston speed. — SCHLICK on the other hand, demonstrates that the weights of the pistons and rods of the first and last engines, which in the five-cylinder arrangement are considerably in excess of that of the middle engine, set up comparatively powerful tripping couples (compare 9) so that ships thus fitted are subject to excessive vibration, and this has also been corroborated by experience.

14) The table on pp. 402 and 403 shews the average working results of all the triples fitted in the sea-going steamers of the North German Lloyd and Hamburg American Lines up to 1892.*) **Table.**

15) The mean piston-speed, col. 5 in the following table, which was about 2.5 m in the latest of the two-cylinder compounds has been raised to an average of 2.8 to 3.0 m for the slower steamers and 4.0 to 4.5 m for the mail steamers against 3.4 for the three-cylinder compounds of the earlier North German Lloyd steamers. Correspondingly the average revolutions have been increased from 60 to 65 for the compounds to about 70 for the mail steamers' triples (col. 16) and 80 for the "Greyhounds". **Increase of Piston-speed.**

16) Col. 4 of the table shews that the working pressure has risen during the eighties by four successive stages from 10 to 12 atmos. The table is accordingly divided into four groups and for each of these the average consumption has been calculated from col. 27 and entered in col. 28. It is thus seen that every rise of pressure was accompanied by a small increase in the economy of coal. It is perhaps useful here to draw attention to the very particular value of the figures relating to coal-consumption in this table. They embrace the faster and the slower ships, voyages in tropical and temperate climates, in calm and stormy weather, with good and bad coals, and skilful as well as untrained firemen, so that the errors arising from all these various circumstances are eliminated by the extensive period of the observations and the great number of voyages. In every case the consumption includes galley and auxiliaries, *whence we may conclude that about 0.75 kg per IHP per hour must be regarded as the smallest consumption attainable at present for triples in average work at sea, not on trial-trips, and exclusive of auxiliaries, galley, &c.* against 0.9 for the best of the compounds, which means an improvement of 17%. **Economy of the Triple.**

*) HAACK & BUSLEY. Die technische Entwickelung des Norddeutschen Lloyds u. s. w. Berlin 1893. pp. 200 & 216.

Table of the Average Working Resu s

Ship's name	Engineers	Year built	Working pressure atmos.	Piston-speed m/sec	Cylinders: Stroke cm	Diameters: H. P. cm	Diameters: M. P. cm	Diameters: L. P. cm	Ratio of volumes.	Power: total average at Sea IHP	Power: per 1 m grate IHP
1	2	3	4	5	6	7	8	9	10	11	12
Ia. Three-cylinder Engines											
"Bayern", "Preussen", "Sachsen"	Vulcan Co., Stettin	1886	10.0	3.25	150.0	91.0	145.0	230.0	1:2.539:6.388	3500	105.4
"Danzig", "Lübeck", "Stettin"	Do.	1886	10.0	2.83	125.0	60.6	100.0	160.0	1:2.723:6.972	1600	114.3
"Gothia"	Richardson, West-Hartlepool	1884	10.2	2.28	91.4	55.9	93.9	147.3	1:2.828:6.950	1140	102.1
"Russia"	Laird, Birkenhead	1889	10.3	3.35	152.4	83.8	132.0	203.1	1:2.481:5.914	2725	72.3
"Aller", "Saale", "Trave"	Fairfield, Glasgow	1886	10.5	3.72	182.9	111.7	177.8	274.3	1:2.531:6.025	7600	102.1
"Francia"	Reiherstieg, Hamburg	1886	10.5	2.35	106.6	55.9	83.8	147.3	1:2.250:6.950	928	91.6
"Ascania", "Colonia"	Wallsend, Newcastle	1887	10.5	2.57	99.0	55.9	88.9	147.3	1:2.531:6.950	1100	109.5
"Galicia"	Clark, Sunderland	1889	10.5	2.34	106.6	60.9	99.0	162.5	1:2.640:7.111	1344	116.0
"Dresden", "München"	Fairfield, Glasgow	1889	10.5	3.38	137.2	76.2	127.0	203.2	1:2.777:7.111	2592	90.[illegible]
"Darmstadt", "Gera", "Karlsruhe", "Oldenburg", "Stuttgart", "Weimar"	Do.	1890	10.5	3.38	137.2	78.7	132.1	210.8	1:2.814:7.168	3000	100.[illegible]
"Dania", "Scandia"	Vulcan Co., Stettin	1889	11.0	3.13	140.0	84.0	135.0	215.0	1:2.583:6.551	3409	113.3
"Kaiser Wilhelm II."	Do.	1889	11.0	3.65	160.0	105.0	170.0	270.0	1:2.621:6.621	6000	109.[illegible]
"Sumatra"	Howaldtswerke, Kiel	1889	11.0	1.97	50.0	36.0	56.5	93.0	1:2.463:6.673	340	100.[illegible]
"Italia"	Armstrong, Newcastle	1889	11.2	3.13	121.9	64.8	104.1	170.1	1:2.586:6.906	1970	133.[illegible]
"Cheruskia", "Markomannia"	Stephenson, Hebburn	1890	11.2	2.45	106.6	64.8	104.1	170.1	1:2.586:6.906	1570	94.[illegible]
"Croatia"	Blohm & Voss, Hamburg	1889	11.2	2.40	100.0	56.0	89.0	147.5	1:2.526:6.935	1030	122.[illegible]
"Helvetia"	Wallsend, Newcastle	1889	11.2	2.51	114.3	60.9	101.6	162.5	1:2.777:7.111	1459	104.[illegible]
"Flandria"	Reiherstieg, Hamburg	1888	11.5	2.49	106.6	57.1	91.4	147.3	1:2.549:6.658	1150	122.3
"Virginia"	Blohm & Voss, Hamburg	1891	11.6	2.79	107.3	64.1	101.6	167.5	1:2.511:6.838	1750	100.1
"Venetia"	Reiherstieg, Hamburg	1891	12.0	2.41	106.6	64.8	104.1	165.7	1:2.586:6.550	1540	108.4
	Averages*)		—	2.86	—	—	—	—	1:2.602:6.769	—	101.[illegible]
Ib. Three-cylinder Engines											
"Augusta Victoria"	Vulcan Co., Stettin	1889	10.5	4.00	160.0	105.0	170.0	270.0	1:2.621:6.612	12000	115.4
"Fürst Bismarck"	Do.	1891	11.0	4.40	160.0	110.0	170.0	270.0	1:2.388:6.002	14150	104.8
"Columbia"	Laird, Birkenhead	1889	11.0	4.16	167.6	104.1	167.6	256.5	1:2.591:6.068	12450	109.2
"Normannia"	Fairfield, Glasgow	1890	11.25	4.91	167.6	101.6	170.1	269.2	1:2.805:7.022	14840	110.[illegible]
	Averages		—	4.37	—	—	—	—	1:2.601:6.426	—	109.8
II. Four-cylinder Engines											
"Hohenzollern", "Hohenstaufen"	Vulcan Co., Stettin	1890	11.0	2.71	125.0	2×56.0	126.0	206.0	1:2.531:6.766	2200	114.6
"Habsburg", "Salier"	Do.	1891	12.0	2.92	125.0	2×48.0	108.0	176.0	1:2.531:6.722	1900	153.3
III. Five-cylinder Engines											
"Lahn"	Fairfield, Glasgow	1887	10.5	4.27	182.9	2×82.6	172.8	2×216.0	1:2.189:6.841	8700	107.2
"Spree", "Havel"	Vulcan Co., Stettin	1890	11.0	4.20	180.0	2×95.0	190.0	2×250.0	1:2.000:6.925	11800	113.4

*) "Sumatra" is omitted as too small.

of 46 Triple-expansion Engines.

Boilers			Screw							Average speed of ship	Coal-consumption				
Grate surface	heating surface	grate surface / heating surface	Average Revolutions	No. of blades	Diameter	pitch mean or uniform	projected surface	Indic. Thrust on projected surface	Slip		hourly per □m of grate	daily	hourly per IHP		
													lbs engl.	kg	Average for same or nearly same working pressure
□m	□m				m	m	□m	kg/□cm	%	Knots	kg	t. engl.			kg
13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28
of Mail and Cargo Steamers.															
33.20	1026.40	1:30.9	65	4	5.500	7.400	7.140	0.510	10.0	14.0	84.3	67.000	1.770	0.800	
14.00	418.00	1:29.8	68	4	4.500	6.150	4.780	0.246	11.4	12.0	89.3	30.000	1.720	0.781	
11.16	345.80	1:30.9	75	4	4.930	5.033	5.984	0.264	10.2	11.0	77.5	20.443	1.674	0.759	
37.70	840.00	1:22.5	66	4	5.638	6.705	7.246	0.422	9.5	13.0	64.9	57.777	1.979	0.898	0.809
74.41	2102.40	1:28.2	64	4	6.750	9.296	10.100	0.626	12.1	17.5	83.7	152.000	1.810	0.820	
10.13	306.86	1:30.3	66	4	4.571	5.029	5.247	0.258	7.2	10.0	73.8	17.673	1.777	0.806	
10.03	327.94	1:32.6	78	4	4.571	4.419	4.273	0.376	11.4	10.0	85.0	20.323	1.710	0.776	
11.60	360.00	1:31.0	66	4	4.876	5.333	5.945	0.310	8.0	10.5	98.9	27.080	1.880	0.853	
29.40	952.63	1:32.4	74	4	5.182	6.096	4.288	0.682	11.1	13.0	68.4	47.000	1.680	0.760	
30.00	993.00	1:33.1	74	4	5.258	6.248	4.864	0.692	13.2	13.0	73.0	53.000	1.610	0.730	0.791
29.28	864.90	1:29.5	67	4	6.000	6.600	8.300	0.460	8.0	13.0	90.6	64.484	1.765	0.800	
54.73	1680.00	1:30.7	68.5	4	6.650	7.700	10.990	0.532	12.3	15.0	90.9	120.000	1.830	0.830	
3.40	129.29	1:38.0	118	4	2.400	2.900	1.700	0.292	10.0	10.0	86.0	7.000	1.900	0.860	
14.19	497.00	1:33.7	77	4	4.876	4.876	6.224	0.420	9.6	11.0	106.0	35.550	1.684	0.764	
16.70	510.00	1:30.5	69	4	5.182	5.334	6.354	0.360	10.0	10.75	70.0	27.488	1.634	0.741	
8.38	260.73	1:31.1	72	4	4.720	4.720	4.752	0.308	7.7	10.25	100.5	19.918	1.805	0.818	
14.00	438.60	1:31.3	66	4	4.876	5.333	5.945	0.341	8.0	10.5	72.0	23.829	1.524	0.691	0.786
9.48	346.36	1:36.5	70	4	4.571	5.029	5.130	0.311	8.5	10.5	94.8	21.236	1.724	0.782	
17.16	520.00	1:30.3	78	4	4.876	4.952	4.874	0.475	12.2	11.0	77.7	32.083	1.711	0.776	
14.20	522.60	1:36.8	68	4	4.876	5.638	5.505	0.373	11.4	11.0	83.5	28.459	1.700	0.770	0.776
—	—	1:31.2	70.0	4	—	—	—	0.419	10.1	—	83.4	—	—	0.787	
of Swift Steamers.															
104.00	3344.00	1:32.1	75	4	5.050	9.550	6.650	0.730	22.4	18.0	100.5	247.000	1.921	0.871	
135.00	4356.00	1:32.2	85	3	5.800	8.500	6.000	0.905	18.8	19.0	89.8	287.000	1.890	0.857	
114.00	3245.00	1:28.4	75	3	5.486	9.448	5.382	0.903	18.5	18.7	107.1	288.000	2.164	0.981	
135.00	4315.00	1:32.0	88	3	5.524	8.150	5.759	1.007	20.0	18.6	94.8	305.500	1.900	0.862	—
—	—	1:31.4	80.7	3	—	—	—	0.886	19.9	—	98.0	—	—	0.893	
of Mail Steamers.															
19.2	640.0	1:33.3	65	4	5.200	6.800	6.800	0.368	9.1	13.0	95.5	4.40	1.840	0.833	
12.6	530.0	1:44.2	70	4	5.100	6.100	6.000	0.386	11.7	12.0	121.5	3.50	1.690	0.767	—
of Swift Steamers															
81.2	2361.1	1:29.8	70	4	6.780	9.140	11.611	0.608	15.0	18.0	92.3	18.00	1.900	0.862	
104.0	3434.0	1:33.0	70	4	6.850	9.500	12.890	0.722	14.0	18.5	97.3	24.30	1.892	0.858	—

§ 47.

Quadruple-expansion Engines.

First Quadruple-expansion Engine.

1) **I. Introduction of the Quadruple-expansion Engine.** The *first* quadruple-expansion engine *ever built* was fitted by ADAMSON*) in 1875 at a cotton mill near Manchester. The working pressure was 8.83 atmos. and there were two cranks 90° apart. The first and second cylinders worked one crank and the third and fourth the other. The dimensions were

Cylinder	I . . .	0.432 m	diameter	and	1.524 m	stroke,
"	II . . .	0.570 "	"	"	" "	"
"	III . . .	0.768 "	"	"	" "	"
"	IV . . .	1.067 "	"	"	" "	"

During a four hours' run the average *IHP* was 540 and the consumption 0.8 kg of coal per horse per hour. The cylinders were not jacketed, but the eduction pipes from II to III and from III to IV were heated with boiler steam not only to prevent condensation but with the avowed object of superheating the working steam.

First Marine Engine.

2) The first quadruple-expansion engine which remained at work *in a ship* was that of the S. S. "County of York" built by the Barrow Shipbuilding Co. in 1884. In the summer of 1886 there were already four steamers with quadruple engines classed in Lloyd's Register and since then many others have followed, so that at the present time there are probably about 100 of these engines partly at work and partly under construction.

Classification.

3) The quadruples hitherto built may best be classified according to the number of their cranks as
   a) two-cranked,
   b) three-cranked,
   c) four-cranked,
   d) five-cranked

Two-cranked Quadruples.

4) **II. In Two-cranked quadruples** the double tandem arrangement with two cranks 90° apart is adopted although it has many inconveniences. If we call the first two cylinders the H.P's. and the second two the L.P's., as is often done, we may distinguish two different designs of these engines.
   a) It is most usual for the two H.P's. and the two L.P's. respectively to form one tandem engine, i. e. that cylinders I and II actuate the fore crank and III and IV the after one.
   b) Another arrangement is more rarely met with, in which the two H.P's. are placed over the two L.P's. so that I and III have the fore crank and II and IV the other.

*) Engineering, Sept. 10, 1875. Quadruple Engines.

c) A third and very peculiar arrangement of the four cylinders (side by side over 2 cranks 180° apart) has been introduced by FLEMING & FERGUSON of Paisley.

5) **a. The first arrangement** was adopted in most of their engines by RANKINE & BLACKMORE of Greenock as well as by FLEMING & FERGUSON at first, also by DENNY of Dumbarton in BROCK'S patent engines, both new and converted from two-cylinder compounds. As to the distribution of the work in the cylinders and upon the cranks of engines of this type the author's only experience has been gained from the S. S. "Kronprinz Friedrich Wilhelm". Her particulars are exhibited below in comparison with those of engines of the second arrangement. RANKINE & BLACKMORE claim that in the engines fitted by them to the S. S. "Myrtle"*) $^1/_4$ of the *IHP* was done in each of the cylinders, but as the diagrams are not published, this assertion cannot be checked. This engine is further stated to have worked as smoothly as a three-cranked triple in consequence of careful balancing and the favourable effect of the compression. **The first arrangement.**

6) **b. The second arrangement** is exemplified in the engines of the steamers "County of York"**) by the Barrow Shipbuilding Co. and "Suez"***) by the Central Marine Engineering Co. of West-Hartlepool, the former being a new engine, the latter a conversion from a two-cylinder compound. Both worked at almost the same pressure and piston-speed and developed about equal powers. As the following comparative table shews, the number of expansions in the smaller cylinders of the "County of York" was about 11 and in the larger ones of the "Suez" almost 16. These data are here attached to those of the "Kronprinz Friedrich Wilhelm". **The second arrangement.**

	Pressure in H.P. cylinder atmos.	Vacuum atmos.	Total ratio of Expansion	*IHP*	Piston-speed m. per sec.	Revolutions
County of York	10.89	0.895	1 : 10.82	983.6	2.06	58
Suez	10.68	0.947	1 : 15.90	985.5	2.04	56
Kronprinz Fr. Wilhelm	10.90	0.855	1 : 12.00	1513.1	7.26	67

7) **c. The third arrangement** was first applied by FLEMING & FERGUSON in the engines of the steam yacht "Skeandhu"†) of 120 *IHP*; those of the cargo steamer "Singapore"††) of 1600 *IHP* are of **The third arrangement.**

*) Engineering. 1887 I. p. 546.
**) Engineering. 1887 I. p. 297.
***) The Engineer 1888 I. p. 162.
†) Engineering. 1888 I. p. 538.
††) Engineering. 1889 I. p. 328.

the same type. The latter is the sixth similar engine at work and there were ten more building in the spring of 1889. In the "Skeandhu" all the cylinders are at the same height above the crankshaft. The piston-rods of I and III are connected by links and a bell-crank of cast steel to the fore crank, II and IV in the same manner to the after one. The apex of the bell crank (which replaces the connecting rod) carries the crank-pin brasses. All the pumps are worked by levers off the forward bell-crank. The two pistons of each pair of cylinders do not work exactly together, but the one has a certain amount of lead with respect to the other, so that there is no dead point. The steam is admitted in the middle of a piston valve and passes thence to cylinder I, on exhausting from this into the steam-chest it is admitted on the *outside* of a second piston-valve on the same rod as the first into cylinder II, and exhausts through the *inside* of this piston-valve into cylinder III. Cylinders III and IV are worked in the same way with two piston valves. All the valves are driven by excentrics and links and those of cylinders III and IV are immediately over the shaft so that they can be worked in the usual way. But the valves of I and II are a considerable distance out of the centre line, on account of which the excentric straps of cylinders III and IV are connected by rods with a small rocking shaft carried in the bed plate of the engine and communicating motion to the link for the valves of I and II. The two links are arranged in a line one before the other and their drag-links are connected to the reversing weigh-shaft. Thus the four cylinders are worked with only two excentrics. The advantages claimed for this engine are — the accessibility ot its cylinders, its shortness fore and aft and its lowness compared with the tandem, which is important in war-ships. As regards spaçe it may here be mentioned that the "Singapore" engine of 1600 *IHP* is but 4.3 m long and only occupies about 18 sq. m. area of engine seat. There are fewer moving parts of this engine than any other quadruple so that it requires less attention and is cheaper to work. The moving weights are balanced by the cranks being opposite to each other and the transmission of the piston-loads through the bell cranks to the shaft is said to take place as if there were four cranks at right angles. The "Skeandhu's" engines are reported to have run very quietly and smoothly at 400 revolutions.

Comparison of two-crank Engines.

8) In order to furnish a comparison of two-cranked engines with others, the following table has been prepared, for the first two engines with as much accuracy as the small scale of the published diagrams permitted and with complete exactness for the third.

Table referring to Two-crank Quadruple-expansion Engines.

		Cylinder I.	Cylinder II.	Cylinder III.	Cylinder IV.	Fore Crank	After Crank	Difference between the two cranks
"County of York"	Range of temperature in °C. .	27	33	29	35	—	—	—
	Initial load in kg	11392	16221	13800	14491	25192	30713	5421
	Mean ind. pressure in atmos.	3.804	2.56	1.146	0.53	—	—	—
	IHP	212	283.6	251	237	463	520.6	57.6
"Suez"	Range of temperature in °C. .	28	37	48	36	—	—	—
	Initial load in kg	14159	22435	31445	17728	45604	40163	5441
	Mean ind. pressure in atmos.	2.84	1.97	1.126	0.474	—	—	—
	IHP	195	252	296	242.5	491	494.5	3.5
"Kronprinz Friedrich Wilhelm"	Range of temperature in °C. .	20	31	25	38	—	—	—
	Initial load in kg	10302	18869	15008	13108	29171	28116	1055
	Mean ind. pressure in atmos.	3.515	2.825	1.176	0.699	—	—	—
	IHP	314.5	510.4	420.0	504.0	824.9	924.0	99.1

From this table it is evident that the equality both of range of temperature and of initial loads in the cylinders leaves much to be desired and the equal distribution of the work on the two cranks was only successfully achieved in the "Suez". No fault can however be found with the distribution in the "County of York" and "Kronprinz Friedrich Wilhelm" as many designers arrange the work on the cranks to increase from forward to aft. It further appears from the combined diagrams the author has constructed for these engines that their economy in the use of steam is only moderate. The "County of York" shews a very considerable steam loss at the "drop" between cylinders I and II which is probably due to the large clearance spaces of the passages and piston-valves which are small and not particularly well designed. In the "Suez" which also has piston-valves for cylinders I and II similar losses occur, although they are not so important. They are however excusable here, as this was not a new engine, but only a conversion from an old compound with the slides placed between the cylinders where it was an object to retain the original excentric gear and link motion. It was therefore a necessity to design the four new cylinders so as to bring in the original positions of the slide-rods. The circumstances of the conversion of the "Kronprinz Friedrich Wilhelm's" old compound engine were exactly similar.

Conversion of Compounds to Quadruples.

9) In converting old compounds into quadruples the first arrangement referred to in 6) has often been selected and BROCK'S Patent relates to it also, as he aims at working two tandem cylinders with one common slide-valve. Engines of this type have been fitted to new steamers, as "Jumna", "Jelunga", and "Lahore". It is probable that the very large unbalanced slide common to the third and fourth cylinders did not work so well in engines of considerable dimensions as in the former smaller ones, so that

the designer was led to introduce four slides, one for each cylinder in place of the original two. This was the arrangement of the "Buenos Ayres"*) (a new engine) as well as the "Kronprinz Friedrich Wilhelm". The probable reason why less is heard of engines of this type now than at the end of the eighties is that the conviction has prevailed that their consumption is no lower than that of three-crank triples of the same size, initial pressure, and number of expansions. It is perhaps even rather higher, as three-stage expansion is certainly more favourable for a pressure of 12 atmos. than four-stage expansion, to say nothing of the superiority in smoothness of working of the three-crank engine over the two-crank, although BROCK states that this difference disappears if the compression in the upper cylinders of the latter is properly managed. The "Kronprinz Friedrich Wilhelm" burns 0.83 kg of coal per horse per hour at sea, so that she is not so economical as the triples quoted on p. 402. This is probably the state of affairs with other engines of this type, as Messrs DENNY'S firm, who have long afforded a brilliant example in the matter of carefully conducted trial-trips, have never published the consumption of these numerous engines both new and converted, which they would assuredly have done if the results had been remarkably good.

Three-crank Quadruples.

10) **III. Three-crank quadruple-expansion engines.** The only engine of this kind which has become at all known**) is that of the steam yacht "Rionnag-na-Mara" built by RANKIN & BLACKMORE of Greenock. But it can not be regarded as a fair example, as there are three H.P. cylinders one placed over each of the cylinders II, III, and IV. This peculiar construction was necessitated by the special requirements of the owner. The only interest attaching to this engine centres in the distribution of work in the cylinders, which came out as follows;

		Cylinder I.	Cylinder II.	Cylinder III.	Cylinder IV.	Crank I.	Crank II.	Crank III.
12 Expansions	Range of Temperature in °C.	17	36	31	37	—	—	—
	Initial load in kg	1016	6638	5521	4732	7654	6537	5748
	Mean ind. pressure in atmos.	4.42	3.67	2.2	0.766	—	—	—
	IHP	33.3	144	164	136	177.3	197.3	169.3
13 Expansions	Range of Temperature in °C.	17	32	37	37	—	—	—
	Initial load in kg	1016	6276	6724	4732	7292	7740	5748
	Mean ind. pressure in atmos.	4.21	3.07	1.76	0.654	—	—	—
	IHP	28.3	108	118	104	136.3	146.3	132.3

*) Engineering 1888 II. p. 415.

**) Engineering 1886 I. p. 361.

It thus appears that she worked rather more favourably with 13 expansions than 12, for the work on the several cranks approached more closely to equality. Nevertheless the ranges of temperature and initial loads varied a great deal and did not agree nearly so well as in good three-crank triples.

11) HARMAN*) of Partick claims in his patent of 1888 for a quadruple engine, to obtain all the advantages of a three-crank shaft and proposes the following arrangements. **Harman's arrangements.**

a) The first three cylinders, viz. the H.P. A^1, the first M.P. B^1, and the second M.P. C^1 are tandemed over three equal L.P's. D^1, D^2, D^3. This plan appears suitable only for very large engines, where subdivision of the L.P. is necessary.

b) The first two cylinders A^1 and B^1 with two equal L.P's. D^1 and D^2 form the outside engines, between which are placed C^1 and C^2 dividing the third stage of the expansion between them. This plan has more prospect of success, although it is necessary to apportion the greater share of the work to the last two stages of the expansion in order to approximately equalize the duty upon the cranks.

c) Two equal H.P. cylinders A^1 and A^2 over two equal L.P's. D^1 and D^2 form the forward and after engines, while the smaller M.P. B^1 over the larger M.P. C^1 make up the middle engine. Here however, having regard again to the equality of work on the cranks, the range of temperature must be kept much smaller in the M.P's. than the other cylinders.

12) As all these plans are open to more or less serious objection, **Superior Design.** in consequence of the difficulty of distributing the four-stage expansion work equally over three-cranks, it is simplest to abandon, as far as possible, the tandem arrangement on account of its inherent drawbacks and to design the three-crank quadruple with one H.P. cylinder over the L.P. and the two M.P's. as independent engines, analogously to the two-crank triple. — DENNY adopted something like this plan in the "Queen Olga"**), but Nos. I and II were tandemed and III and IV separate. The H.P. could be disconnected, enabling the other three to work as a triple at lower power.

13) **IV. Four-crank quadruple-expansion Engines** have hitherto been designed according to the three following plans. **Four-crank Quadruples.**

a) WIGHAM RICHARDSON & Co.'s,

b) SCHLICK'S,

c) CRAMP & SONS'.

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*) Engineering. 1888 I. p. 361.

**) The Marine Engineer. 1893/94. p. 531.

14) a. WIGHAM RICHARDSON & Co. of Newcastle-on-Tyne have made quadruples for several years. The Hamburg S. S. "Ramses" exemplifies their design. The four cylinders are arranged in the following order from forward III, I, II, IV. The cranks of I and IV as well as of II and III are 180° apart while those of I and II are at right angles. As crank No I leads, the others II, IV, and III follow at intervals of 90°. The working pressure is 14 atmos., the engine runs smoothly, and the ship is free from vibration at her low speed of 10 knots. The consumption is stated to be only 0.7 kg per horse per hour at sea. The Fairfield Co. also fitted a quadruple to the "Tantallon Castle"*) in 1894, the cylinders being in the order from forward I, II, IV, III, but nothing is stated as to the crank-arrangement.

Schlick's Quadruple.

15) b. SCHLICK**) proposes the following plan. The order of the cylinders from forward is 1, 3, 4, 2 (Plate 11, Fig. 10), or when weight is no object so that the weights for pistons 3 and 4, as well as for 1 and 2, can be made equal, their order is 1, 4, 3, 2, to shorten the steam pipe between 2 and 3. When the piston weights are equal it is best to place the cranks of 3 and 4 120° apart as the difference between the greatest and least twisting moment then becomes the smallest, in fact only about half as great as when these two cranks are at right angles. At the same time the drop of pressure between the several cylinders is reduced.

Determining the positions of the Cranks.

16) SCHLICK determines the positions of cranks I and II thus. We will assume the cranks III and IV to enclose an angle of 120° (Plate 11, Fig. 11), the distances from centre to centre of cylinders to be equal, and the weights of the moving parts $P_1 = P_2$ and $P_3 = P_4$ the last two of which are known. The effect of inertia in No. IV engine would be completely eliminated if the cranks of I and II could be placed exactly opposite No. IV crank and if $P_1 + P_2$ were $= P_4$. But as the weights P_1 and P_2 act at different distances from P_4, the weight P_2 must be made $= {}^2/_3\, P_4$ and $P_1 = {}^1/_3\, P_4$. In a similar way the effect of inertia in No. III engine is to be overcome by putting opposite to its crank $P_2 = {}^1/_3\, P_3$ and $P_1 = {}^2/_3\, P_3$. If now we construct the resultants of the opposing forces acting at cranks IV and III, their directions give the positions of cranks I and II and their lengths the magnitudes of the weights P_1 and P_2 respectively. In the present special case to which the above assumptions apply, cranks I and II enclose an acute angle. If neither the

*) Engineering. 1894 II. p. 668.

**) Zeitschrift des Vereins deutscher Ingenieure 1894. p. 1096.

weights nor the distances from centre to centre of cylinder are equal, and if the valve gear is also to be balanced, then the most favourable angle between P_3 and P_4 is no longer 120^0 and that between I and II approaches 90^0 more closely.

17) c. The firm of CRAMP & SONS of Philadelphia adopted two four-crank quadruples in their new fast steamer "St. Louis" of the American Line, each having six cylinders for a total *IHP* of 19000. One H.P. forms a tandem with one L.P. and the order from forward is A_1 with D_1, B, C, A_2 with D_2. The two forward cranks are opposite each other and have equal weights of pistons and rods, the third crank is at right angles to the second and the fourth opposite the third, because the weights for these two cranks are also equal. It must be left to experience to shew whether this design will materially reduce vibration. **Cramp's arrangement.**

18) **V. A five-crank quadruple** was fitted by HERRESHOFF*) in the U. S. torpedo-boat "Cushing". The diameters are 28.6, 40.6, 57.1, 57.1, and 57.1 cm, those of the third, fourth, and fifth cylinders being the same; the last two form the L.P. cylinder divided in two and take the fourth stage of the expansion. The forward engine is the H.P. and the others follow in order so that the two L.P's. aft complete the series. Each crank encloses an angle of 144^0 with the next, making the total angle from the H.P. to the after L.P. 720^0. It was hoped that this arrangement would greatly tend to equalize the transmission of the loads to the crank-shaft and to diminish the vibration always met with in such light fast boats at full speed; "Cushing" went 22 Knots. Unfortunately ISHERWOOD'S**) very elaborate report on her trials makes no reference to vibration, so no judgment can be formed as to the success of HERRESHOFF'S crank-arrangement. **Five-crank Quadruples.**

19) **VI. An economical superiority of the quadruples** built up to the end of 1894, as compared with triples has not been demonstrated, but they appear to be about as much better than compounds as these are than low-pressure engines. This latter point is of importance because when boilers have to be renewed, old compounds can be so simply converted into quadruples. But trustworthy accounts of the consumption of such converted engines are as rare as those of new quadruples, although they have been described at length in the technical press. A few particulars of consumptions known to the author are given below. **Coal-consumption of Quadruples.**

*) Journal of the American Society of Naval Engineers 1890 p. 215.

**) Journal of the American Society of Naval Engineers 1892 p. 9.

Table of the Coal-consumption of Quadruple-Expansion Engines.

No.	Steamer Type	Steamer Name	Year built	Builder	IHP	Boiler pressure atmos.	Consumption per IHP per hour	Remarks
1	Torpedo-boat	"Cushing"	1890	Herreshoff	1720	17.58	1.01	Forced-draught trial
2	Mail-steamer	"Kronprinz Friedrich Wilhelm"	1887	Denny	1750	12.00	0.83	Average at sea
3	Torpedo-boat	Yard No. 450.	1891	Schichau	1714	15.00	0.80	Forced-draught trial
4	Cargo steamer	"Ramses"	1893	Wigham, Richardson & Co.	1400	14.00	0.70	First voyage
5	Steam yacht	"Myrtle"	1887	Rankin & Blackmore	400	12.65	0.55	Trial-trip
6	Steam yacht	"Rionnag-na Mara"	1886	Rankin & Blackmore	544	12.65	0.51	Trial-trip
7	Steam yacht	"Skeandhu"	1888	Fleming & Ferguson	120	12.65	0.51	Trial-trip
8	Cargo steamer	"Singapore"	1889	Fleming & Ferguson	1600	11.60	0.51	Trial-trip

Trustworthiness of the Table.

20) The consumption of the first three engines is undoubtedly correct. That of the fourth will probably increase a little on later voyages when the boilers are no longer quite clean. Those of Nos. 5 to 8 reported in English periodicals as quoted *are given much too small, for they are considerably below what is theoretically possible.* In § 36, 13 it is demonstrated that the smallest theoretical consumption attainable with a perfect engine at 14 atmos. working pressure and 9 expansions comes to 0.51 kg. At the working pressure of 12.65 or even only 11.6 atmos. and assuming the factor of evaporation to have been 10, such a low consumption is absolutely impossible with these engines, which were of course not perfect. Reports of this kind are therefore only calculated to mislead the unprofessional reader.

Poor Results of the Quadruple hitherto.

21) **VII. Prospects of the general adoption of the quadruple.** The quadruples hitherto constructed have not shewn any commercial improvement on the triple for the very obvious reason that their range of temperature was too small for four-stage expansion. There is no prospect of an actual advantage of the quadruple over the triple until we get to initial pressures of more than 15 atmospheres, i. e. temperatures of over 200° C. and a total range of temperature of at least 130° C., and even then the advantage will be much less than that of the triple over the compound.

Objections to the Quadruple.

22) The prospects of a more general adoption of the quadruple are for the present limited by

a) the fact that the increase of power is not in proportion to the increase of pressure,
b) the increasing steam-losses,
c) the greater expense both for capital and up-keep,
d) the present want of properly adapted boilers.

23) **a. The gain of work** and the other advantages explained on p. 378 due to an increase of working pressure, become smaller, the higher the pressure from which the increase starts. On calculating the work produced by 1 kg of steam in expanding down to 0.5 atmos. absolute as is usual in most engines, it will be found that the work done at 20 atmos. initial pressure is to the work done at 12 atmos. initial pressure approximately as 26 : to 23, or in other words that from an increase of pressure of 8 atmos. only 11% increase of work can be even theoretically looked for, which will of course be less in practice. **Small gain from increased Pressure.**

24) **b. The steam-losses,** although they are diminished by the reduction of the range of temperature in each cylinder of a quadruple, are on the other hand increased by friction and cooling on the walls of the longer passages and the three receivers over what they would be in a triple of the same size, so that the saving of steam due to the four-stage expansion is partly sacrificed again. **Increased Steam-losses.**

25) **c. The first-cost of the engine** is raised by the extra expense of patterns for four cylinders, pistons, slides, steam-chests, and covers, and for the fourth crank and rods. The addition of a fourth engine of course increases the cost of maintenance and in most cases, especially in steamers remaining long away from their home port, it is necessary to augment the engine-room staff. **Increased Cost.**

26) **d. The present want of properly adapted boilers** is one of the most important obstacles, because a rise of working pressure above 15 atmos. with the present types of boilers (apart from water-tube ones) increases their weight and cost to such an extent that it is questionable whether the improved economy to be expected from higher pressure is anything like an equivalent. Another objection, and on the score of danger, is the high temperature of the steam. It is 158° C. at 5 atmos., 183° C. at 10 atmos., but as much as 214° C. at 20 atmos. As the boiler plates which transmit the heat of the fires to the water have a rather higher temperature than the steam, and mild steel must be regarded as untrustworthy at a blue heat which occurs at about 245° C., it is clear that at 20 atmos. working pressure the difference between the temperature of the steam and that at which the boiler plates begin to change their character is only half as great as at 10 atmos. and only about one third as great as at 5 atmos. **Want of proper Boilers.**

Table of Trial-trip Results of

№	Name of Ship	System of Engine	Type	Builder	Year of Trial	Mean pressure in atmos.: Boiler p_κ	Main steam-pipe at engine	H.P. initial p	Ratio of clearance to volume swept m	Ratio of cut-off ε or ε_g
1	2	3	4	5	6	7	8	9	10	11
									I. Single-Expansion	
1	Dispatch-boat "Grille"	1 Twin-engine jet-condenser no jackets no superheater	Horizontal Trunk-Engine	Penn, Greenwich	1858	1.33	1.10	0.85	0.070	0.600
2	Cruiser "Medusa"	"	"	"	1865	1.33	1.20	0.80	0.075	0.660
3	Iron-clad "Kronprinz"	1 Twin engine jet-condenser jackets superheater	"	"	1868	1.66	1.51	1.20	0.080	0.600
4	Gun-boat "Nautilus"	1 Twin engine jackets surface-condenser no superheater	"	Oderwerke, Grabow	1872	1.99	1.85	1.40	0.150	0.350
5	Iron-clad "Kaiser"	1 Twin-engine jackets surface-condenser superheater	"	Penn, Greenwich	1874	2.00	1.86	1.44	0.095	0.400
6	Cruiser "Freya"	1 Three-crank engine jackets surface condenser superheater	"	Germania Co., Berlin	1876	1.92	1.80	1.35	0.110	0.400
7	School-ship "Blücher"	"	"	"	1879	2.07	1.90	1.53	0.090	0.375
									II. Single-Expansion	
8	Armoured gun-boat "Basilisk"	2 Twin-engines jackets surface condenser no superheater	Diagonal direct acting	WeserCo., Bremen	1879	3.92	3.80	3.44	0.085	0.150
9	Tender "Ulan"	1 Twin-engine jackets surface condenser no superheater	Inverted direct acting	Oderwerke, Grabow	1879	5.18	5.00	4.60	0.145	0.300
										III. Woolf
10	Iron-clad "Oldenburg"	2 Coupled engines jackets surface condenser	Horizontal direct acting	Vulcan Co., Stettin	1886	5.14	5.00	4.70	HD 0.087 ND 0.065	0.222
11	Cruiser "Prinzess Wilhelm"	"	"	Germania Co., Berlin	1889	7.00	6.90	6.70	HD 0.132 ND 0.081	0.147

the Engines of 20 German War-ships.

Mean indicated pressure or reduced mean pressure in atmos. p_i	Back-pressure in Condenser in atmos. a	Revolutions per Minute N	Piston speed in m per Second c	IHP	Useful steam per IHP per hour measured from diagrams in kg D_{IHP_n}	Gross steam per IHP per hour calculated from coal consumption with factor of evaporation in col. 22 kg D_{IHP}	Ratio of useful to gross steam ξ	Coal per IHP per hour kg	Coal per □m of grate per hour kg	Factor of evaporation assumed from experience kg	Mean speed of ship in Knots	Duration of trial in hours
12	13	14	15	16	17	18	19	20	21	22	23	24
Low-pressure Engines.												
1.54	0.20	128.2	1.95	765	12.00	19.14	0.62	2.20	122.0	8.7	13.43	3
1.56	0.06	75.0	1.90	775	11.60	17.85	0.65	2.10	120.0	8.5	10.54	4
1.45	0.11	68.6	2.75	4835	11.25	16.53	0.68	1.90	137.0	8.7	14.33	3
1.22	0.17	103.8	1.63	496	11.30	18.60	0.60	2.14	118.6	8.7	10.61	4
1.35	0.03	77.0	3.12	7695	9.70	13.92	0.70	1.60	140.0	8.7	14.41	3
1.20	0.18	78.0	2.04	2118	10.40	15.56	0.67	1.79	120.7	8.7	14.88	3
1.30	0.09	94.6	2.71	2990	9.20	12.80	0.72	1.47	135.4	8.7	13.86	3
High-pressure Engines.												
1.96	0.05	116.0	1.74	700	8.80	11.34	0.77	1.40	96.0	8.1	10.10	4
2.44	0.45	102.1	1.70	781	8.30	10.53	0.79	1.30	84.3	8.1	12.20	4
Engines.												
1.93	0.10	92.8	2.66	4004	7.41	8.67	0.85	1.07	93.5	8.1	13.40	6
2.34	0.10	100.9	2.89	9241	7.30	8.83	0.83	1.09	157.1	8.1	18.00	6

№	Name of Ship	System of Engine	Type	Builder	Year of Trial	Mean pressure in atmos. Boiler p_x	Main steam-pipe at engine	H.P. initial p	Ratio of clearance to volume swept m	Ratio of cut-off ε or ε_g
1	2	3	4	5	6	7	8	9	10	11
									IV. Two-cylinder	
12	Dispatch-boat "Blitz"	2 Engines jackets surface condenser	Horizontal direct acting	Germania Co., Berlin	1883	5.05	4.85	4.65	HP 0.165 LP 0.131	0.208
13	Dispatch-boat "Greif"	"	"	"	1887	6.72	6.50	6.31	HP 0.117 LP 0.070	0.139
									V. Three-cylinder	
14	Gun-boat "Möwe"	1 Engine jackets surface condenser	"	Schichau, Elbing	1880	5.61	5.40	5.10	HP 0.090 LP 0.070	0.210
15	Cruiser "Olga"	"	"	Vulcan Co., Stettin	1881	5.08	4.80	4.40	HP 0.130 LP 0.090	0.140
									VI. Triple-	
16	Dispatch-boat "Wacht"	2 Engines only H. P. jacketed surface condenser	Diagonal	Weser Co., Bremen	1888	10.0	9.80	9.60	HP 0.122 MP 0.103 LP 0.072	0.114
17	Cruiser "Falke"	2 Engines no jackets surface condenser	Horizontal	Imperial Dockyard Kiel	1891	12.00	11.80	11.65	HP 0.120 MP 0.100 LP 0.080	0.092
18	Iron-clad "Beowulf"	"	Inverted direct-acting	Weser Co., Bremen	1892	11.90	11.70	11.50	HP 0.121 MP 0.100 LP 0.080	0.117
19	Iron-clad "Wörth"	"	"	Germania Co., Berlin	1894	12.10	11.80	11.70	HP 0.160 MP 0.130 LP 0.090	0.105
20	Cruiser "Gefion"	"	"	Schichau, Elbing	1895	12.10	11.50	11.40	HP 0.170 MP 0.140 LP 0.100	0.121

Mean indicated pressure or reduced mean pressure in atmos.	Back-pressure in Condenser in atmos.	Revolutions per Minute	Piston-speed in m per Second		Useful steam per IHP per hour measured from diagrams in kg	Gross steam per IHP per hour calculated from coal consumption with factor of evaporation in col. 22 kg	Ratio of useful to gross steam	Coal per IHP per hour	Coal per □m of grate per hour	Factor of evaporation assumed from experience	Mean speed of ship in Knots	Duration of trial in hours
p_i	a	N	c	IHP	D_{IHP_n}	D_{IHP}	ξ	kg	kg	kg		
12	13	14	15	16	17	18	19	20	21	22	23	24
Compound Engines.												
1.82	0.17	146.0	3.21	2808	8.27	10.36	0.80	1.28	130.7	8.1	15.71	3
2.16	0.10	105.0	3.02	5432	7.82	9.88	0.81	1.22	150.1	8.1	19.14	6
Compound Engines.												
1.88	0.12	130.0	2.86	875	7.47	9.35	0.80	1.14	103.1	8.2	11.71	4
2.08	0.08	106.5	2.05	2394	7.50	9.30	0.80	1.15	98.0	8.1	13.92	4
expansion Engines.												
2.35	0.23	156.0	3.12	4134	6.89	8.44	0.81	1.03	283.8	8.2	20.14	6
2.41	0.11	132.0	3.30	2878	6.72	7.33	0.91	0.894	123.9	8.2	16.92	6
2.53	0.19	144.0	3.80	4871	6.83	7.69	0.88	0.938	224.5	8.2	15.11	6
2.86	0.36	109.5	3.65	10228	6.23	7.13	0.87	0.880	143.4	8.1	17.02	6
2.43	0.16	139.0	4.08	9533	5.88	6.32	0.93	0.785	93.8	8.1	20.18	6

Further Development.

27) The untiring progress of engineering will certainly before long produce boilers which are fit to stand higher working pressures and there will then be a corresponding transition to more numerous stages of expansion. In the present state of practice however as shewn above, there are many and various obstacles in the way and the advantage to be expected is not attractive enough.

28) **VIII. Concluding remarks on the selection of a type of engine.** *The two-cylinder compound* working up to 7 atmos. pressure still remains the most suitable for inland navigation as the short trips of small river steamers do not bring out the economy of the triple like long voyages at sea. Besides, the undeniable saving of 17 % of fuel (see p. 401) has to be set against a much higher percentage of increased capital and working expenses than in sea-going steamers. The conditions are similar for paddle mail-boats on short runs. — *The three-cylinder triple* up to 12 atmos. retains its pre-eminence for sea-going screw steamers of such low speeds as not to be subject to vibration and whose power does not exceed the limits of a *single* L.P. cylinder less than 2.5 m in diameter. It is sufficiently economical for present circumstances, has a satisfactory useful effect, and runs uniformly enough, nor does it require too long an engine space nor too large a staff. — *The four-cylinder quadruple* of at least 14 atmos. working pressure with its four cranks arranged on SCHLICK'S plan (see 16, p. 410) is likely to be the engine of the near future for all swift sea-going screw steamers, because the four cranks check vibration. If in any case four cranks are adopted without the necessity of dividing the L.P. cylinder of a triple into two, then the quadruple is certainly preferable to the triple, because at 14 atmos. pressure it is likely to be about 6 % more economical than the triple at 10 to 12 atmos. (see "Ramses" in the table p. 412).

Table.

29) The preceding table exhibits the proportionate economy of the different systems of machinery and the progress of marine engineering during the last forty years. As no feed-water measurements were taken on any of the trials, the total steam-consumption is calculated from the coals as measured, assuming 1 kg of coal to evaporate the weight of water given in Col. 22, as confirmed by experience for the respective type of boiler. The figures thus derived for the total steam-consumption in Col. 18 and the fraction ξ calculated from Cols. 22 and 18, expressing the ratio of the useful to the total steam-consumption (see § 31,30), have therefore only a relative value. — The factors of evaporation were taken at 8.7 for good box boilers, 8.1 for

good return-tube cylindrical boilers, and 8.2 for good Navy-type and locomotive boilers.

30) As on the trial-trips reported in the table the object was to get the highest power out of the engines and economy was quite a secondary consideration, the consumptions both of coal and steam were greater than on ordinary runs at more favourable cut-offs. Besides this there is the circumstances that mercantile engines both compound and triple always have smaller H.P. cylinders than war-ships' engines of equal power, which sufficiently explains their smaller consumptions as quoted on pp. 393, 394, and 402. **Lower values for mercantile engines.**

Eighth Division.

Calculation of the Dimensions of the Cylinders.

§ 48.

Calculation for Single-expansion Engines.

Object.

1) **I. Determination of the Diameter.** Although single-expansion engines are no longer used as main engines, scarcely even for boats, but only for auxiliary machinery, the calculation for them will nevertheless be gone through, because those for compounds and multiple-expansion engines can be so conveniently based upon it.

Definitions.

2) The following definitions apply to all the succeeding paragraphs of this division.

I. Dimensions of Cylinders.

D diameter of cylinder of a single-expansion engine,
D_h „ „ H.P. „ ,
D_m „ „ M.P. „ ,
D_n „ „ L.P. „ ,
d „ „ piston-rod or trunk,
O effective piston-area,
v volume of H.P. cylinder,
V „ „ L.P., or (§ 51, 6) of one L.P., or (§ 52, 11) of M.P. cylinder,
R volume of the receivers.

II. Piston-travels.

H stroke of piston,
c piston-travel per sec.,
m fraction of stroke travelled by H.P. piston at the moment of cut-off in L.P. of a two-cylinder compound, or M.P. of a triple,
m' uncompleted fraction of stroke of H.P. piston of a three-cylinder compound at moment of cut-off in L.P. No. I,
m'' uncompleted fraction of stroke of piston of L.P. No. II of a three-cylinder compound at moment of cut-off in L.P. No. I.

III. Pressures.

p initial pressure in all cylinders of a single-expansion engine or H.P. of a multiple-expansion engine,

p_r receiver pressure immediately after H.P. exhaust,

p'_r „ „ at beginning of stroke in L.P.,

p''_r „ „ immediately before H.P. release,

p_μ „ „ „ „ M.P. cut-off,

p_n „ „ „ „ L.P. „ „ at more than half-stroke,

p^0_n receiver pressure immediately after L.P. cut-off where this occurs before half stroke,

p'_n receiver pressure between L.P. cut-off and H.P. release,

p''_n „ „ during L.P. admission,

p'''_n pressure in L.P. cylinder during expansion,

p_1 receiver pressure at cut-off of both L.P.'s of a three-cylinder compound,

p_I terminal pressure in L.P. No. I of a three-cylinder compound,

p_{II} terminal pressure in L.P. No. II of a three-cylinder compound,

p'_I initial pressure in L.P. No. I of a three-cylinder compound,

p'_{II} „ „ „ „ „ II „ „ „ „ „

a condenser pressure.

IV. Cut-off ratios.

ε cut-off ratio in all cylinders of a single expansion engine and H.P. of a multiple expansion engine,

ε_m cut-off ratio in M.P.,

ε_n „ „ „ „ L.P.,

ε'_n and ε''_n special cut-off ratios in the L.P's.,

ε_g total cut-off ratio of multiple-expansion engines.

V. Other weights, pressures &c.

m mass of moving parts,

P weight of „ „ ,

F load on piston due to their inertia,

q pressure on „ „ „ „ „ per □cm,

g acceleration of gravity = 9.81 m,

w mean peripheral velocity of crank-pin circle,

N revolutions per minute,

l length of connecting-rod,

r arm of crank.

3) In determining the diameter of the cylinder the following method is the most to be commended, if it is proposed to investigate the probable uniformity of working and ratio of maximum to mean twisting moment, which may be absolutely necessary for multiple **Course of the calculation.**

expansion engines in particularly swift vessels; that is to say we have to

a) sketch an indicator card such as is likely to be obtained from the projected engine,
b) determine the mean pressure p_o,
c) assume the stroke H,
d) decide upon N, the revolutions per min.,
e) calculate the diameter D of the cylinder.

Theoretical Indicator-diagram.

4) a. **Sketch of indicator-diagram.** Referring to Plate 12, Fig. 1, AB is the stroke H assumed according to 8), and AC the initial pressure p on any convenient scale. As we are here considering a low-pressure engine, with 2 atmos. working pressure, and therefore an absolute boiler pressure p_x of 3 atmos., p would be by Eq. 109, p. 235, $=$ 2.5 atmos., but it is better to take it rather lower, to be on the safe side, say 2.4 atmos. Make $CD = h$, the travel of the piston during admission. Here it is best to employ the most favourable cut-off ratio ε_v (p. 238). But if the engine is to be particularly light, and therefore the cylinders as small as possible, the cut-off ratio must be greater, whereby of course the economy suffers. Through D draw DF parallel to AC, then the rectangle $ACDF$ represents the work during admission. To find the expansion work, MARIOTTE'S curve (§ 18, 29) must be drawn from D to G (§ 6, 16), for which, neglecting the clearance (compare § 18, 44), A is taken as the pole. From the zero line AB which represents absolute vacuum set off a equal to the back pressure which is here taken at 0.15 atmos. (see p. 240) and draw MN, then the area $MCDGN$ is the *theoretical* indicator diagram.

The exhaust line.

5) To complete the diagram the exhaust and compression lines must now be drawn in. The falling pressure in the cylinder during the exhaust and the subsequent decrease of the back pressure can be represented with sufficient accuracy by a parabola constructed as follows. In Fig. 3, Plate 12, $MPRSN$ is a theoretical diagram, in which VW is the atmospheric line, MN the zero line, XY the back-pressure line, RS the expansion line constructed according to § 6, 16, and A the point of release. From O the centre of VW and OW as radius strike the arc CWD, the length of which is determined by the chord CD perpendicular to VW, divide this arc into any number of equal parts and draw through the points thus given a series of parallels to CD. Thereupon draw AF parallel to VW and from centre B with any radius BE which must be greater than AB, strike the arc EF. Through F and B draw a circle from centre G on AB, divide EB into the same number of equal

parts as the arc CD, strike an arc from each of these points of division through B and draw through the intersections of these arcs with the circle FB a series of parallels to VW. The intersections of these horizontal parallels with the vertical parallels drawn through the points of division of the circle CWD give the points I, II, III &c. of the required parabola.

6) The compression curve beyond KX is often represented by the parabolic arc from B to IV. This method is based upon the following reasoning. The mixture of steam and water left in the cylinder at the beginning of the compression has most probably the same composition as that existing at the same moment in the condenser. The pressure of the vapour is a function of the temperature of the cylinder and this may be assumed to rise during the compression at the same rate as the temperature of the condenser falls after the exhaust is completed. — But in the case of an engine with an unusual amount of compression the compression line must be drawn as an adiabatic according to § 18, 12. The Compression.

7) b. **Mean pressure** p_0. The mean pressure is determined from the diagram by § 30, 13^b and at the same time calculated by Eq. 120, β being selected according to the values given on p. 248. If these two methods do not give the same mean pressure the diagram must be corrected by the help of some cards actually taken from engines of the same type and having similar valve-gear, by altering the admission, exhaust, and compression lines to the eye, until the mean pressures obtained by both processes are fairly equal. In all the following calculations the actual mean pressure p_0 is used as found by Eq. 120. Mean pressure p_0.

8) c. **The stroke** H is always made as great as the available space will allow, bearing in mind that l the length of connecting-rod must be at least $4\,r$. Long-stroke engines work more uniformly and smoothly than those with short stroke, especially at high piston-speeds. On the other hand, a long stroke of course increases the weight. This is particularly noticeable in large engines, whereas the influence of the stroke upon the weight vanishes in small engines, because the weight of pistons, cross-heads, and connecting-rod heads exceeds the weight of the rods themselves. — The length of stroke varies very much and depends altogether upon the purpose of the ship and the type of engine. Paddle engines have in general the longest stroke, often as much as 2 m and above. Horizontal engines of war-ships have strokes of 0.4 m in small gun-boats up to 1.37 m in heavy iron-clads. The inverted engines of recent war-ships are always of shorter stroke than those of merchant Assuming the stroke.

ships because they have to be got in below the armoured deck. While in the small inverted engines of war-ships, say torpedo-boats, the stroke generally varies between 0.35 and 0.5 m and seldom exceeds 1 m in the large engines of iron-clads and cruisers, it rises in fast mail-steamers as high as 1.83 m (see the table p. 402). As a guide in fixing upon a stroke for a proposed new engine, use can be made of the various tables in this book, a list of which is given at the end of the last volume.

Choice of number of Revolutions.

9) d. **The number of revolutions per minute** N of course depends upon the piston-speed c when once the stroke has been fixed. This speed is about 2.5 m in the older war-ships and recent slow cargo boats, it rises to 3 and 3.5 m in mail-steamers, 3.5 to 4 in modern fast war-ships, 4.5 in swift steamers (see table p. 402) and 5 and above in torpedo-boats. If c and H are given, we have

$$N = \frac{60\,c}{2\,H}$$

The propeller must be arranged to suit these revolutions and the ship's speed, and it is advisable for swift ships to take into account what is said in § 46, 6 & 7. The revolutions fall with increasing size of engine, as the following table referring to merchant steamers shews. War-ships have a higher number of revolutions in order to bring out their high piston-speed at their short stroke. Engines of iron-clads of 5000 *IHP* work at 100 to 120 revs. those of fast cruisers up to 140 even, and torpedo-boat engines at 300 to 400.

Table of Mean Revolutions of Screw Steamers.

1	2	1	2	1	2	1	2
IHP	Revolutions per Minute	*IHP*	Revolutions per Minute	*IHP*	Revolutions per Minute	*IHP*	Revolutions per Minute
200	150—160	1200	100—105	2700	75—80	5000	70—75
400	130—140	1500	95—100	3000		6000	
600	120—130	1800	90—95	3500		7000	
800	110—120	2100	85—90	4000		8000	
1000	105—110	2400	80—85	4500		10000	

Calculation of the Cylinder Diameter.

10) e. **The diameter of the cylinder** D in cm is determined from the intended *IHP* as follows. By Eq. 126 the theoretical gross power of the engine is

$$IHP = 0.000349\ (D^2 - d^2)\,H\,N\,p_0\,x$$

whence follows $D^2 - d^2 = \dfrac{IHP}{0.000349\,H\,N\,p_0\,x}$.

The diameter of the piston-rod is provisionally neglected but in the days of trunk-engines the diameter of the trunk had to be taken into account from the first and was usually put at 0.75 H. In determining this diameter afterwards for good, it was fixed by the greatest angle of oscillation of the connecting rod and so arranged that there was never less than 5 to 10 mm clearance between the rod and the inner edge of the mouth of the trunk. For a single-expansion engine with one piston rod and no tail-rod we have

$$D = \sqrt{\frac{IHP}{0.000349\, H\, N\, p_0\, x}}\ \text{cm} \quad \dots\dots\dots \quad (214)$$

11) *Numerical example.* As there are no data referring to a recent single-expansion engine, the particulars of the trunk-engine of the German torpedo-school ship "Blücher" designed about 1875 are here selected. The data at that time were $IHP = 2500$, $x = 3$, $H = 0.86$ m. The piston-speed c had to be 2.5 m, whence **Example.**

$$N = \frac{60\, c}{2H} = \frac{60 \times 2.5}{2 \times 0.86} = 87 \text{ revolutions.}$$

The absolute boiler-pressure p_x was taken at 3 atmos. as was usual for low-pressure engines and consequently the most advantageous cut-off ratio with surface condenser and superheater (p. 239) was $\varepsilon_v = 0.35$. The actual mean pressure by Eq. 120 is then

$$p_0 = \beta\, \zeta\, p_x$$

From the table on p. 245 the value of ζ corresponding to $\varepsilon = 0.35$ is found to be $\zeta = 0.7174$. The coefficient β depends upon the valve-gear and with cam-motion the smallest, i. e. the safest value is $\beta = 0.56$ (see § 29, 21 p. 247), then

$$p_0 = 0.56 \times 0.7174 \times 3$$

$$p_0 = 1.20 \text{ atmos.}$$

We can now substitute in the equation evolved in 10)

$$D^2 - d^2 = \frac{IHP}{0.000349\, H\, N\, p_0\, x} = \frac{2500}{0.000349 \times 0.86 \times 87 \times 1.2 \times 3} = 26594\ \square\text{cm}$$

$$D = \sqrt{26594 + d^2}$$

The required diameter of trunk was found to be $d = 73.4$ cm, therefore $D = \sqrt{26594 + 73.4^2} = \sqrt{31981}$

$$D = 178.7 \text{ cm (it was actually made 178.2 cm).}$$

12) **II. Investigation of the degree of uniformity during the design.** The degree of uniformity of a single-expansion engine depends, as shewn in § 42, 4 and 5, on the range of the tangential loads. **How the desired Degree of uniformity affects the calculation.**

The tangential load is influenced

a) by the weight of the moving parts,

b) by the steam pressure accelerating their motion,

c) by the length of the connecting rod.

In the case of the "Blücher" the investigation of the degree of uniformity was gone into in the first instance on strictly theoretical lines i. e. with an imaginary indicator diagram and an infinitely long connecting rod and afterwards with the actual trial-trip cards and the real length of connecting rod in order to obtain a comparison between the design and the practical result.

Weight of Rods. 13) a. **The weight of the moving parts,** as piston and rod, connecting rod and cross-head, as well as any pump buckets with their rods which may be driven direct off the piston, affects the working of the engine as follows. When the piston is on the centre its velocity is zero but rises during one half of the stroke till it equals the peripheral velocity of the crank-pin circle. To impart this change of velocity to the above mentioned parts a certain amount of work is required and has to be performed by the steam. Thus in the first half of the stroke, the crank does not get the benefit of the whole of the forward pressure on the piston but only the excess of it over the pressure required to accelerate the motion of the moving parts. As in the second half of the stroke their motion is again retarded to zero by the action of the crank, they here deliver to it the energy stored up in them during the first half stroke, so that during this period the steam pressure acting to its full extent on the piston is increased by the pressure due to the above delivery of energy. It therefore comes about that during a whole stroke although the whole work of the steam is transmitted to the crank this does not take place as it appears to do by the indicator diagram which only registers the steam pressure on the piston, but according to another law influenced by the moving masses. For the purposes of a design and in the absence of trustworthy data from actual engines of a similar character, the following weights of rods expressed as pressures per cm of piston area will serve as a guide. They represent the weight of piston and rod, cross head, connecting rod, and half the crank for each cylinder.

0.17	to	0.18	kg	for	single-expansion		low-pressure	engines,	
0.20	"	0.22	"	"	"	"	high-	"	"
0.40	"	0.50	"	"	the	H.P.	cylinder	of	compounds,
0.15	"	0.17	"	"	"	L.P.	"	"	2-cylinder compounds,
0.22	"	0.25	"	"	"	L.P.	"	"	3- " "
0.20	"	0.30	"	"	"	H.P.	"	"	ordinary triples,
0.12	"	0.15	"	"	"	M.P.	"	"	" "
0.07	"	0.10	"	"	"	L.P.	"	"	" "
0.15	"	0.20	"	"	"	H.P.	"	"	specially light-triples (torpedo-vessels),

0.06 to 0.10 kg for the M.P. cylinder of specially light-triples (torpedo-vessels),
0.04 „ 0,06 „ „ „ L.P. „ „ specially light-triples (torpedo-vessels).

14) **b. The load due to the acceleration of the masses** F in horizontal engines like the "Blücher's" is inversely proportional to the cosine of the angle β made by the crank with the horizontal, i. e. it is greatest at the dead-point and zero at half stroke. In vertical engines β is the angle made by the crank with the vertical. At the beginning of the stroke the acceleration load is **Pressure due to Acceleration.**

$$F = \frac{P w^2}{g r} \quad \ldots\ldots\ldots\ldots\ldots \quad (215)^{*)}$$

For the middle cylinder of the engine under consideration (compare 24) we have $P = 4104.5$ kg; $r = 0.43$ m; $N = 87$, consequently

$$w = \frac{2 r \pi N}{60} = \frac{2 \times 0.43 \times 3.1415 \times 87}{60} = 3.9 \text{ m}$$

and
$$F = \frac{4104.5 \times 3.9^2}{9.81 \times 0.43} = 14793.7 \text{ kg}$$

If O is the effective piston-area the pressure per sq. cm. at the beginning of the stroke due to acceleration is

$$q = \frac{F}{O}$$

or as here $O = \frac{\pi}{4}(D^2 - d^2) = \frac{\pi}{4}(178.2^2 - 73.4^2) = 20708$ sq. cm

$$q = \frac{14793.7}{20708} kg = 0.707 \ kg.$$

For any position of the crank in the first half of the stroke, we have by the preceding

$$q = \frac{F}{O} \cos\beta \text{ or here } q = 0.707 \cos\beta \text{ kg} \quad \ldots\ldots \quad (216)$$

This value is negative and to be subtracted from the corresponding ordinate of the theoretical or projected indicator diagram. For any position of the crank during the second half of the stroke q is positive and must be added.

15) **c. With an infinitely long connecting rod,** which is the provisional assumption here, the acceleration pressure diminishes steadily up to half stroke and then gradually increases again, the ends of a set of ordinates representing it being in a straight line. **Infinite Connecting rod.**

*) The derivation of this and the following formulæ will not be gone into as they lie beyond the scope of this work; the reader is referred to the original by J. F. Radinger. — „Ueber Dampfmaschinen mit hoher Kolbengeschwindigkeit." Edition III. Vienna 1892. p. 9.

Classification of the diagrams.

16) The projected indicator diagram (Fig. 1, Pl. 12) being completed, the following diagrams are to be constructed from it, in order to arrive at the degree of uniformity.

α) diagram of acceleration pressures,
β) „ „ horizontal (or vertical) loads,
γ) „ „ tangential loads,
δ) „ „ combined tangential loads,
ε) „ „ mean tangential load.

Theoretical Diagram of Acceleration pressures.

17) α. This diagram is constructed by setting off the initial acceleration pressure $\frac{F}{O}$ found as in 14) as the ordinate CK on the line CL of the steam pressure diagram Fig. 1, Pl. 12, and through the point of bisection S of CL drawing a straight line KSR which then exhibits the effect of the acceleration pressures. The area KCS is to be subtracted from and the area SLR to be added to the area of the steam pressure diagram. These two areas KCS and SLR together make up the required diagram of acceleration pressures.

Theoretical Diagrams of horizontal or vertical pressures.

18) β. **The diagram of horizontal or vertical pressures respectively** results when the above-described addition and subtraction of the acceleration diagram has been carried ont. To do this the steam pressure diagram is divided into 10 equal strips and the several ordinates diminished or increased by the corresponding acceleration ordinates. Fig. 5, Pl. 12 shews the horizontal pressure for every position of the crank of a horizontal engine, i. e. the *horizontal pressure diagrams* $MKVWN$ for the out-stroke. For the instroke we obtain the diagram $N\,W_1\;V_1\,K_1\,M$, symmetrical to the above. In the case of vertical engines the weight of the rods per sq cm of piston area is to be subtracted for the up-stroke and added for the down-stroke to produce the *vertical pressure diagram* (see 36).

Construction of the theoretical tangential load.

19) The tangental loads are constructed from the horizontal or vertical pressures, as the case may be, and as Fig. 13, Pl. 11 shews, for infinite connecting rod, the tangential load is $P \sin \beta$, β being the angle of inclination of the crank with the centre line of cylinder. In this figure $OF = r \sin \beta$ and $DJ = P \sin \beta$, so that

$$r : OF = P : DJ \text{ or}$$
$$r : OF = P : P \sin \beta.$$

By the help of these ratios the tangential loads are easily found from the horizontal (or vertical) loads by the following method which avoids the use of the sine. Join A with F which corresponds to any position of the crank, lay off from O towards

A the respective horizontal (or vertical) load and draw from the point A_1 thus obtained, a parallel to AF, then the distance OF_1 thus cut off on OE represents the tangential load for this position of the crank.

20) γ. **The tangential load diagram** is obtained from the horizontal load diagram (which expression will in future be used for the sake of brevity as signifying either horizontal or vertical load diagram) by drawing a semicircle about it representing the path of crank-pin, Fig. 1, Pl. 13 which is afterwards extended as shewn in Fig. 3, Pl. 13 and the tangential loads found as above erected as ordinates at the corresponding points. The tangential load diagram refers to *one* crank only, the full lines to the out-stroke and the dotted lines to the in-stroke. **Theoretical extended diagram of tangential loads.**

21) δ. **The combined tangential load diagram** for the out-stroke of all three pistons in the three-crank engine (of the "Blücher") here considered, is constructed by drawing as Fig. 5, Pl. 13 shews, the separate tangential load diagrams of the three cranks one over the other in positions corresponding to the angles between the cranks, and adding the three ordinates at each spot for the ordinates of the combined curve. **Theoretical combined tangential load diagram.**

22) As the above process of expanding the semicircle is rather laborious, it is usual in England and France to construct the diagram in a circular or "clock" form by laying off the several loads from the centre of the crank pin circle direct upon the radii at their respective positions, as shewn for one cylinder in Fig. 3, Pl. 14. The combined diagram Fig. 5 is obtained by adding together the loads for all three cranks as before. This last diagram, drawn to half the scale of the others, shews the tangent loads during a complete revolution, whereas Fig. 5, Pl. 13 only refers to the out-stroke of the piston. **Theoretical circular diagram of tangential loads.**

23) ε. **The diagram of mean tangent load.** The area of the tangent load diagram Fig. 3, Pl. 13 is equal to the area of the horizontal load diagram Fig. 1, Pl. 13 and also to that of the steam pressure diagram Fig. 1, Pl. 12, for, neglecting friction, all three represent the work done during one revolution of the crank, that is the product of force and travel. If a straight line be drawn in the diagram Fig. 5, Pl. 13 cutting the boundary curve so that the areas above and below this straight line are equal, it will be the line of mean load. In a similar manner the circle of mean load may be drawn in Fig. 5, Pl. 14. The closer the highest and lowest points are to this line (or circle), the more uniformly the engine works. In the present case the maximum tangent load is to the mean as 1.087 : 1. **Theoretical Diagram of mean tangent load.**

Inertia of the rods.

24) *The energy stored up in the moving parts*

$$\frac{m v^2}{2} = L \text{ is} = \frac{P v^2}{2 g} \text{ mkg} \quad . \ . \ . \ . \ . \ . \ . \ . \ (217)$$

because

$$m = \frac{P}{g}$$

In this engine, the weight P for the middle cylinder is composed of

the weight of the piston with both trunks . 3072.0 kg,

" " " " connecting rod . . . . . 1032.5 "

which has to be increased for the other two cylinders by

the weight of the air-pump rod and bucket . 203.5 kg.

" " " " feed and bilge pump rams . 79.0 "

The energy of the rods for the middle cylinder at half stroke (where the velocity c of the piston = the peripheral velocity w of the crank-pin circle, see 13) which is delivered up during the second half of the stroke is therefore

$$L = \frac{4104.5 \times 3.9^2}{2 \times 9.81} = 3182 \text{ mkg.}$$

This energy is expended during a half a stroke or $\frac{60}{2 \times 2 \times N} =$ $\frac{60}{2 \times 2 \times 87} = 0.17$ seconds, so that it amounts to

$$L = \frac{3182}{0.17 \times 75} = 250 \text{ HP},$$

and for all three cylinders to 750 HP, if the small weight of the parts of air, bilge, and feed pumps is neglected. The quantities given in § 42,6 are calculated in this manner.

Investigation of the degree of uniformity of the engine itself.

25) **III. Investigation of the degree of uniformity of a completed engine.** The process of constructing the diagrams from the trial-trip cards is just the same as before; the discrepancies arise only from

a) the influence of the *actual* indicator card upon the diagram of steam-pressures, and

b) the effect of the *finite* length of the connecting rod upon the other diagrams.

Actual steam-pressure Diagram.

26) **a. The steam-pressure diagram** is constructed from the indicator cards of both ends of the cylinder by measuring the pressures from the top of one card to the bottom of the other, i. e. from the forward pressure to the back pressure at each point. Such a diagram, as illustrated by Fig. 2, Pl. 12, gives the pressure actually transmitted by the piston at every moment. At the point U the driving pressure equals the back-pressure and from here to the end of the stroke the work in the cylinder is negative as it is done to the steam during the compression. If this diagram be applied to the line MN, so that the area representing

positive work is above the line and the negative portion below it, we obtain the actual steam-pressure diagram $MCDUGN$ (Fig. 4, Pl. 12) for the out-stroke and $NLD_1\,U_1\,G_1\,M$ for the in-stroke. The negative energy UGN and $U_1\,G_1\,M$ is to be afterwards subtracted from that of the rods.

27) **b. The finite connecting rod,** as compared with an infinite one, gives rise to **Influence of the finite connecting rod.**

α) different velocities of piston in the first and second halves of the stroke,

β) a different distribution of the acceleration pressure,

γ) a change of the tangent pressures.

28) *α. The difference between the piston velocities* in the first and second halves of the stroke is a peculiarity of the mechanism of the crank. While the crank moves from the dead-point A a quarter of a revolution to C (Fig. 12, Pl. 11), the piston travels from a to c, i. e. the distance oc beyond half stroke, it therefore moves faster in the back half of the cylinder than in the front half. This irregularity in the motion of the piston is the more pronounced the shorter the connecting rod is. In order to determine the tangent loads with a finite connecting rod it is therefore necessary to find the exact position of the piston for every angle of the crank (Fig. 2, Pl. 13). **Influence of the difference in the velocity of piston.**

29) *β. The acceleration pressure with a finite connecting rod,* for any angle β of the crank with the centre line is, where $\frac{r}{l}$ is the ratio of crank-arm to length of connecting rod, **Diagram of actual acceleration pressure.**

$$q = \frac{F}{O}\left(\cos\beta \pm \frac{r}{l}\cos 2\beta\right) \text{ atmos.} \quad \ldots\ldots\ldots \quad (218)^*)$$

the plus sign referring to the out-stroke, the minus to the in-stroke. At the dead-point

$$q_1 = \frac{F}{O}\left(1 \pm \frac{r}{l}\right)$$

To construct the acceleration-pressure diagram, the steam-pressure diagram is divided into a number of equal parts, the crank-angle β for each division laid down, and the corresponding value of q worked ont. To facilitate this calculation OTTO H. MÜLLER jun. **) gives the following table for the expression in the bracket (the diagram being divided into the usual 10 parts) for the various values of $\frac{r}{l}$; the figures are of course to be multiplied by $\frac{F}{O}$.

*) See note to Eq. 215.

**) Zeitschrift des Vereins deutscher Ingenieure 1883. p. 282.

Ordinate		$\frac{r}{l} = \frac{1}{4}$			$\frac{r}{l} = \frac{1}{5}$			$\frac{r}{l} = \frac{1}{6}$	
0		1.250			1.200			1.167	
1		0.941			0.920			0.840	
2	Out-stroke	0.664		Out-stroke	0.639		Out-stroke	0.633	
3		0.409			0.379			0.376	
4		0.129			0.126			0.134	
5		—0.101			—0.091			—0.078	
6		—0.318			—0.300			—0.284	
7		—0.471	In-stroke		—0.465	In-stroke		—0.458	In-stroke
8		—0.610			—0.613			—0.608	
9		—0.715			—0.717			—0.742	
10		—0.750			—0.800			—0.833	

The curve $K_1\, S_1\, R_1$ drawn through the ends of these ordinates is shewn in Fig. 1, Pl. 12 for comparison with the line KSR. It cuts the line CL before the half-stroke in S_1 and the distance $S_1\, S$ corresponds to oc in Fig. 12, Pl. 11 (see 28). The curve of acceleration pressures as drawn refers to the crank-end (bottom), that for the cover-end (top) is given in dotted lines in Figs. 4 and 6, Pl. 12. — A very simple process for drawing the acceleration-curve is communicated in the periodical referred to below.*) In the same place RITTERSHAUS gives a construction of the acceleration-curve of the cross-head and piston respectively. KIRSCH**) has published another simpler method of getting out the acceleration-pressure curve which gives specially good information as to the important question of the acceleration at the dead points, and VAES***) has appended to this a still more handy process for determining the accelerations of the piston. But in practice the above table is of the greatest use, because it attains the object most rapidly. — The horizontal pressure diagram now follows analogously to what is said in 18) by combining the steam-pressure diagram with the acceleration-pressure diagram, as drawn in Fig. 6, Pl. 12 and Fig. 2, Pl. 13.

Construction of the actual tangent load.

30) γ. *The change in the tangent load* consists in this, that it is no longer $P \sin \beta$ as with an infinite connecting rod, but that, as explained in § 78,3, it becomes $\frac{P \sin (\alpha + \beta)}{\cos \alpha}$. As however the same ratio exists here between the thrust along the connecting

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*) L. PINZGER. Zeitschrift des Vereines deutscher Ingenieure 1883. p. 282.

**) KIRSCH. Ibid. 1890. p. 1320.

***) F. L. VAES. Ibid. 1891 p. 129.

rod $\frac{P}{\cos\,\alpha}$ and the tangent load $\frac{P \sin\,(\alpha+\beta)}{\cos\,\beta}$ on the one hand, and between the crank arm r and the distance $O\,K$ on the other (see Fig. 13, Pl. 11), as was shewn in 19) to exist with the infinite connecting rod, the tangent load can easily be got out. Having found the corresponding position of crank $O\,G$ for a given position of piston, produce the centre line of connecting rod to K and draw $A\,K$; it is now only necessary to lay off the respective horizontal load, taken from the diagram Fig. 2, Pl. 13, from O to A_1 and lay a parallel to $A\,K$ through A_1 which cuts off on $O\,K$ the distance $O\,K_1$ representing the tangent load.

31) In constructing the diagram of tangent loads it is necessary, as done in Fig. 4, Pl. 13, to mark off upon the developed crank-pin circle the exact positions of the crank corresponding to the several positions of the piston, and to set up from these the ordinates determined by the process described in 30). The diagram for the in-stroke differs considerably from the other in consequence of the finite length of connecting rod. The combined tangent load diagram for all three cylinders (Fig. 6) is produced in exactly the same way as Fig. 5, except of course that the change in the crank positions must be taken into account. Here again the clock diagram can be employed, Figs. 4 and 6, Pl. 14. The latter, drawn to half the scale of the other diagrams, again exhibits the tangent loads during a revolution whereas Fig. 6, Pl. 13 only shews them for one stroke, the out-stroke of piston I. If absolute accuracy is desired it is necessary with a two or three-cranked engine to get out the tangent loads for both strokes of each cylinder from the indicator cards and afterwards combine them, instead of merely repeating the diagram of cylinder I three times over as has been done here for the "Blücher". **Diagrams of actual tangent loads.**

32) For engines with cylinders of equal diameters the tangent load diagram forms at the same time the twisting moment diagram. § 49,5 explains how the twisting moments of multiple-expansion engines are got out. **Diagrams of twisting moments.**

33) In the tangent load diagram Fig. 6, Pl. 13, which takes account of all the defects in the actual engine, the ratio of maximum to mean twisting moment is 1.176 : 1, so that the difference between this ratio and that of the diagram Fig. 5 which is 1.087 : 1, is only 0.089. As in engines of anything like good design the difference between the actual and theoretical ratio of maximum to mean twisting moment is only small, it is common in practice to regard the connecting rod as infinitely long. **Neglecting the effect of the finite connecting rod.**

Compensating for the inequalities.

34) The inequalities, represented by the areas between the line of mean tangent-load and those portions of the load curve which project above it and fall below it are to a certain extent taken up by the flywheel in stationary engines, but as at sea a flywheel is inadmissible, *balance weights on the cranks* are often fitted instead. (See § 97.)

Balance weights for horizontal engines.

35) They are of advantage in horizontal engines also as they oppose the vibrations of the engine caused by the variations in the thrust between the cylinder-cover and the main bearings. At the dead point the steam presses equally against the cylinder-cover and piston. While the load on the cover is not influenced by mass, only that part of the load on the piston is transmitted to the crank which is not absorbed by the inertia of the piston and rods. The difference between the cylinder-cover load and the crank-pin load, viz. the acceleration-load, is manifested as an unbalanced horizontal force and tends to shift the whole engine in the opposite direction to that of the motion of the piston, because the cylinder-cover load is the greater. With a back balance on the crank of a weight equal to that of the other moving parts, the centrifugal force of the balance weight is resolved into a horizontal and a vertical component. The former is equal and opposed to the acceleration load and therefore prevents the vibrations which would otherwise be occasioned by the unbalanced horizontal force, the latter is taken up by the bed plate and engine-seating and has no influence upon the working of the engine.

Balance weights for vertical Engines.

36) It therefore follows that in vertical engines balance weights set up horizontal forces which may become of greater danger to the stability of the engine than its want of uniformity of working when the moving parts are left unbalanced (compare § 46,5). If it is desired to investigate the non-uniformity due to unbalanced weights in a vertical engine, the diagram of horizontal pressures (here they are of course vertical pressures, see 18) must be taken and the ordinates for the down-stroke increased by the weight $\frac{P}{O}$ of these parts per □cm of piston area, and those for the up-stroke diminished by the same amount. The tangent load diagram is then to be constructed from the corrected horizontal pressure diagram in the same manner as described in 19).

37) The following table shews the calculated relation between the maximum and mean tangent loads for single-expansion engines at various cut-off ratios.

Table of the ratio: $\dfrac{\text{Maximum Tangent load.}}{\text{Mean Tangent load.}}$

Kind of Engine.	Cut-off ratio	Maximum Tangent load / Mean Tangent load
Single-Expansion Engine with 1 crank	0.2	2.625
„ „ „ „ „	0.4	2.125
„ „ „ „ „	0.6	1.835
„ „ „ „ „	0.8	1.698
„ „ „ „ 2 cranks at 90°	0.1	1.872
„ „ „ „ „ „ „	0.2	1.616
„ „ „ „ „ „ „	0.3	1.415
„ „ „ „ „ „ „	0.4	1.298
„ „ „ „ „ „ „	0.5	1.256
„ „ „ „ „ „ „	0.6	1.270
„ „ „ „ „ „ „	0.7	1.329
„ „ „ „ „ „ „	0.8	1.357
„ „ „ „ 3 „ „ 120°	0.35	1.176
Two-cylinder Compound Engine	see Table on page 464	
Three-cylinder „ „	„ „ „ „ 393	
Triple-expansion Engine	„ page 507	

§ 49.

Calculation for Woolf Engines.

1) **I. Determination of the cylinder diameters.** The WOOLF engine, like the single-expansion engine, is now scarcely ever adopted in new vessels. The dimensions of both are calculated in the same manner, because the theoretical performance of the steam remains identical whether it expands in one, two, or more successive cylinders. The diameter D_n of the L.P. cylinder is therefore determined by the method numerically exemplified in § 48, 10 and 11, as if the L.P. engine had to do the whole *IHP* by itself at a cut-off ratio ε_{g_v}, the most favourable one (see § 27), with steam taken direct from the boiler at the absolute pressure p_x. The first step is therefore to decide upon the best cut-off. **Calculation of the diar. of L.P. cylinder.**

2) **The most favourable cut-off ratio** (see § 27,5) is determined for WOOLF condensing engines, as for marine engines in general, from the succeeding tables. **Most favourable Cut-off.**

Table of the most favourable cut-off ratio for Woolf Engines by Hrabák. *)

Absolute Initial Pressure p in atmos.	4	5	6	7	8	9	10
Total ratio of cut-off at moderate Expansion (Terminal Pressure 0.6 atmos.) ε_{g_v}	0.150	0.120	0.100	0.086	0.075	0.067	0.060
Total ratio of cut-off at high Expansion (Terminal Pressure 0.4 atmos.), ε_{g_v}	0.100	0.080	0.067	0.057	0.050	0.044	0.040

EMERY based the following *provisional* formula, as he calls it, for calculating the best cut-off ratio upon his trials of the steamers "Bache", "Rush", "Dallas", "Dexter", and "Gallatin", pp. 85 to 88. The results it gives are for ordinary single-expansion engines, too *high*, for large high-class single-expansion engines, *about correct*, and for superior compounds rather *too low*, but will nevertheless serve as a guide. The formula is

$$\frac{1}{\varepsilon_{g_v}} = \frac{p_x + 37}{22} \quad \ldots\ldots\ldots\ldots\ldots \quad (219)$$

p_x being the working pressure of the boiler in ℔s per sq. in. engl. EMERY compiled the following table from the results of his trials and it agrees pretty closely with the above formula.

Table of the most favourable cut-off ratio by Emery.

Working pressure p_x	0.35	0.70	1.00	2.80	4.22	5.62	7.00 atmos.
Expansions $\frac{1}{\varepsilon_{g_v}}$	1.9	2.1	2.8	3.5	4.4	5.3	6.2

Ratio of H.P. to L.P. cylinder.

3) **Cylinder-ratio.** Having fixed upon ε_{g_v} and D_n we must next settle the ratio of the L.P. to the H.P. cylinder. This question may be approached from the following points of view:

a) GRASHOF'S, when the range of piston-load is to be as small as possible and therefore the power as great as possible, then

$$\frac{V}{v} = 0.85 \sqrt{\frac{1}{\varepsilon_{g_v}}} \quad \ldots\ldots\ldots\ldots\ldots \quad (220)$$

b) HRABÁK'S, when the *IHP* on each crank is to be the same, then

$$\frac{V}{v} = 0.9 \sqrt{\frac{1}{\varepsilon_{g_v}}} \quad \ldots\ldots\ldots\ldots\ldots \quad (221)$$

*) J. HRABÁK. Hilfsbuch für Dampfmaschinentechniker. Edit. II. Berlin 1891. Part II p. 106.

c) WERNER'S, when the maximum load and therefore the stresses on the rods (see § 45,6) are to be as small as possible, then

$$\frac{V}{v} = \sqrt{\frac{1}{\varepsilon_{g_v}}} \quad \ldots \ldots \ldots \ldots \quad (222)$$

d) RANKINE'S, when both pistons are to develop equal powers with the lowest stresses on the rods, then, approximately

$$\frac{V}{v} = \sqrt[3]{\left(\frac{1}{\varepsilon_{g_v}}\right)^2} \quad \ldots \ldots \ldots \ldots \quad (223)$$

For marine engines RANKINE'S is preferable to all other formulæ, as it gives the smallest H.P. cylinder. In practice the cylinder ratio $\frac{V}{v}$ is very often made dependent on the working pressure p_x and for mercantile engines the following relations are usual,

p_x under		3 atmos.	working	pressure	. . .	$\frac{V}{v}$	$= 3{,}0$
,, from	3 to 4	,,	,,	,,	. . .	,,	$= 4.0$
,, ,,	5 ,, 6	,,	,,	,,	. . .	,,	$= 4.5$
,, ,,	6 ,, 7	,,	,,	,,	. . .	,,	$= 5.0$

For war-ships the ratio is generally taken lower, in order to get larger H.P. cylinders. (See § 52,5).

4) **The Diameter of H.P. cylinder** D_h is calculated from that of the L.P. D_n and the cylinder ratio $\frac{V}{v}$ when, as is usual in marine engines, both cylinders have the same stroke: H.P. diameter and cut-off.

$$D_h = D_n \sqrt{\frac{v}{V}} \quad \ldots \ldots \ldots \ldots \quad (224)$$

Finally, the cut-off ratio in the H.P.

$$\varepsilon = \frac{\varepsilon_{g_v} V}{v}$$

This completes the principal dimensions of a WOOLF engine.

5) **II. Investigation of the degree of uniformity.** In order to ascertain the distribution of work in the cylinders and the degree of uniformity of running of a proposed WOOLF engine, an ideal indicator diagram for the whole engine must be constructed. From this the diagrams of steam-pressure, acceleration pressure, horizontal (or vertical), and tangent loads for the cylinders separately are got out as described for single-expansion engines in the preceding §. Instead of the combined diagram of tangent pressures, which forms a diagram of twisting moments in the case of an engine with cylinders of equal diameter only, for a WOOLF engine or a compound the twisting moment diagram must be The twisting moment diagram.

specially constructed. The twisting moment for any point of the crank-pin circle is the product of the tangent pressure, the piston area, and the arm of the crank. If the cylinders have the same stroke, as with rare exceptions is the case in marine engines, the diagram of twisting moments can be easily constructed from the separate tangent pressure diagrams by multiplying the ordinates of that belonging to the L.P. cylinder by the cylinder-ratio $\frac{V}{v}$ and afterwards combining this enlarged diagram with that of the H.P. cylinder in the usual way. In the following pages therefore, the design of the ideal indicator diagram only will be discussed.

Design of the ideal indicator diagram.

6) **III. Construction of the theoretical indicator diagrams.** *The ideal indicator diagram,* Fig. 7, Pl. 15 is drawn as described in § 48 with the following alterations; AB is the stroke, AC the initial pressure in the H.P. cylinder, CD the cut-off (in the H.P.). If the clearance is to be taken into account in constructing the expansion curves, which however is unusual (§ 18,44) and is *not* done here, then in order to find the pole O, the clearance of the H.P. cylinder must be regarded as a cylinder of the same diameter as the L.P. cylinder, the height of which, expressed as a fraction of H, gives the distance OA. The rest of the construction is the same as before. The resulting diagram $ACDGB$ is that of a single-cylinder expansion engine of the dimensions of the L.P. cylinder and of the same IHP as the proposed engine and so forms the basis of the further calculation.

Combined diagrams of actual Woolf engines.

7) *The indicator diagrams of actual Woolf engines* are often combined for the purpose of investigating the losses of effect attending the passage of the steam from the H.P. to the L.P. cylinder. By RANKINE'S method*) (compare § 18,39) the H.P. diagram $CDEF$ (Fig. 1, Pl. 15) is applied to the perpendicular AC so that AC represents, to scale, the absolute initial pressure and the L.P. diagram KPJ is placed beneath so as to bring JL equal to the back pressure in the condenser. In order to make a comparison with the ideal diagram, the correct ratio between the cut-off volume in the H.P. and the terminal volume of the expanded steam in the L.P. must be expressed in the figure. For this reason the length of the L.P. diagram must be increased in the ratio of $\frac{V}{v}$ or $\frac{AB}{AL}$. This is done by drawing a number of parallels to AB through JK and making the distance $sq = \frac{V}{v} sr$,

*) W. J. M. RANKINE. On the working of steam in compound engines. Miscellaneous scientific papers. London 1881. p. 455.

whereupon the L.P. diagram $PKGJ$ results. If the WOOLF engine has two L.P. cylinders, they are to be regarded as a single L.P. of double the volume of each of them. When the pole O has been found as explained in 5) the expansion line is constructed as dotted in Fig. 1, Pl. 15, and the losses of effect are then expressed by the shaded areas as shewn.

8) *The construction of the separate diagrams for the cylinders of a proposed Woolf engine* from the ideal diagram will be next described. **Design of the separate diagrams of a Woolf engine.**

a) for a WOOLF engine (i. e. without receiver),

b) „ „ receiver compound.

Although the former case is without immediate practical importance as there are scarcely any such engines left and all are now built with receivers, still the consideration of it is of so much interest that it could not be omitted. It may be mentioned that the calculation of the several pressures with which the diagrams are to be drawn is done much more simply by SCHRÖTER'S graphic process as exemplied for a three-cylinder compound in § 51,28.

9) *The designing of ideal combined diagrams for Woolf engines (i. e. without receivers)* stands in a very simple connection with RANKINE'S method of combining actual indicator diagrams (see Fig. 1, Pl. 15). If AL Fig. 7, Pl. 15, represents the volume of the H.P. cylinder, i. e. $\frac{AB}{AL} = \frac{V}{v}$, the perpendicular LK is the boundary of the L.P. diagram which has the same length as the H.P. one and it is only necessary to proceed as if the short L.P. diagram were to be re-constructed out of the long one. Through LK parallels are accordingly to be drawn to AB and on AB $\frac{sr}{sq}$ is to be made $= \frac{AB}{AL}$, thus producing the curve KP which separates the two diagrams so that $CDKP$ is the H.P. and $KGNMP$ the L.P. one. The curve KP can also be easily found by computing its ordinates. These ordinates, as FQ, represent the back-pressure in the H.P. cylinder and the forward pressure in the L.P. and are calculated as follows. Expressing the volume FL by x, the whole volume v_1 of steam acting upon the L.P. piston in its position F is equal to the volume $v - x$ remaining in the H.P. cylinder plus the volume $x\frac{V}{v}$ expanded so far in the L.P. **Design of combined diagrams for Woolf engines.**

$$v_1 = (v - x) + x\frac{V}{v}$$

The pressure p_1 or the length of the ordinate FQ for this place is then by MARIOTTE'S law

$$p \varepsilon v = p_1 \left[(v - x) + x \frac{V}{v}\right]$$

$$p_1 = \frac{p \varepsilon v}{(v - x) + x \frac{V}{v}}$$

$$p_1 = \frac{p v}{\frac{v}{\varepsilon} + x\left(\frac{1}{\varepsilon_g} - \frac{1}{\varepsilon}\right)} \quad \ldots\ldots\ldots\ldots \quad (225)$$

According to RANKINE, WOOLF engines work most satisfactorily when the curve KP divides the diagram into two equal areas as then both pistons do equal work.

Receiver pressure immediately after the H.P. release p_r

10) *The construction of combined diagrams for receiver compounds* is carried out as follows. In the diagram Fig. 2, Pl. 15, the steam enters the H.P. cylinder at the initial pressure $AC = p$, is cut-off at D, and expands to E the end of the stroke. At this point connection is made with the receiver and the volume v of steam at the pressure $p\,\varepsilon$ passes into it. This changes the pressure $BR = p_r''$ of the steam present in the receiver to $p_r = LK$, for

$$p_r (v + R) = p_r'' R + p \varepsilon v$$

$$p_r = \frac{p_r'' R + p \varepsilon v}{v + R} \quad \ldots\ldots\ldots\ldots \quad (226)$$

As is here shewn in the diagram, the receiver pressure rises at the moment of the H.P. release and the pressure of the steam leaving that cylinder must necessarily fall as exhibited by the line EK.

Receiver pressure immediately before the L.P. cut-off p_n

11) The pressure p_r is also the initial pressure in the L.P. cylinder and accordingly $AP = p_r = LK$. During admission in the L.P. the volume AQ corresponding to the cut-off ratio ε_n passes into it, and the pressure p_r in the receiver falls in proportion to the increase of volume produced by the forward movement of the L.P. piston. For instance the volume of steam enclosed between the H.P. and L.P. pistons at H, the point of L.P. cut-off is

$$V_n = (1 - \varepsilon_n) v + R + \varepsilon_n V$$

and its pressure $p_n = QH$ is by the preceding

$$p_n V_n = p_r'' R + p \varepsilon v$$

$$p_n = \frac{p_r'' R + p \varepsilon v}{(1 - \varepsilon_n) v + R + \varepsilon_n V} \quad \ldots\ldots\ldots \quad (227)$$

By substituting in this equation various values of ε_n, the pressures corresponding to them, i. e. the ordinates of the curve PH can be determined.

12) Up to the L.P. cut-off the *driving* pressures in the L.P. and the *back* pressures in the H.P. are identical, so that KS can easily be drawn after PH is completed, for it is only necessary, as is at once evident from the preceding, to reduce the abscissa rs in the ratio of the volumes of the cylinders, cut off the distance AP thus found from KL and at the point thus determined to erect as ordinate the pressure corresponding to the abscissa rs. For instance let the pressure at rs be yz, then $\frac{qw}{rs}$ is made $= \frac{AL}{AB}$ and at the point w the ordinate yz erected, giving a point on KS. The pressure TS at the last point of this curve is of course $= QH$. **Receiver pressure during the L.P. admission $p'_{n'}$**

13) The steam in the L.P. cylinder expands from the point H to its terminal pressure $p\,\varepsilon_g = BG$ and volume V. The ordinates of the expansion curve HG are determined as follows. If for instance the volume $AZ = \varepsilon'''_n V$ is enclosed in the L.P. cylinder, then the corresponding pressure $UZ = p'''_r$ is obtained from the initial pressure and volume by the equation. **Pressure in L.P. cylinder during expansion p'''_n**

$$p'''_n \varepsilon'''_n V = \varepsilon_n V p_n$$

$$p'''_n = \frac{\varepsilon_n V p_n}{\varepsilon'''_n V} = \frac{\varepsilon_n}{\varepsilon'''_n} p_n$$

in which the value of p_n taken from Eq. 227 is to be substituted.

14) While the steam in the L.P. cylinder expands from H to G, the steam remaining in the receiver is compressed by the H.P. piston during the completion of its stroke, causing the receiver pressure to rise again from $p''_n = TS$ to $p''_r = AF - BR$. We have now to determine the curve of receiver pressure HR, shewn dotted in the diagram; employing the following equation which represents the initial and terminal state of the steam in the L.P. cylinder **Receiver pressure immediately before the H.P. release p''_r**

$$p_n \varepsilon_n V = p\,\varepsilon_g V$$

$$\left[\frac{p''_r R + p\,\varepsilon\, v}{(1-\varepsilon_n)v + R + \varepsilon_n V}\right]\varepsilon_n V = p\,\varepsilon_g V$$

$$\frac{p''_r R}{v} + p\,\varepsilon = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[(1-\varepsilon_n) + \frac{R}{v} + \frac{\varepsilon_n V}{v}\right]$$

$$p''_r = \frac{p\,\varepsilon_g v}{\varepsilon_n R}\left[(1-\varepsilon_n) + \frac{R}{v} + \frac{\varepsilon_n V}{v}\right] - \frac{p\,\varepsilon\, v}{R}$$

Substituting in the last term $\varepsilon v = \varepsilon_g V$, we get

$$p''_r = p\,\varepsilon_g\left[\frac{\left(\frac{1}{\varepsilon_n} - 1\right)v}{R} + \frac{1}{\varepsilon_n} + \frac{\varepsilon_n V v}{\varepsilon_n R v} - \frac{V}{R}\right]$$

$$p_r'' = p\,\varepsilon_g \left[\frac{\left(\frac{1}{\varepsilon_n} - 1\right) v}{R} + \frac{1}{\varepsilon_n} \right] \quad \ldots\ldots\ldots \quad (228)$$

Inserting different values for ε_n in this formula, as $AZ = \varepsilon_n'''$, &c, the corresponding receiver pressures ZW are found. Finally the receiver pressure curve HR must be shortened in the direction of SF which is done as explained in 12) for the curves PH and KS. The diagram can now be completed.

Losses of effect.

15) **IV. Losses of effect.** If the receiver pressure p_r'' is less than the pressure $p\,\varepsilon$ of the steam exhausting from the H.P. cylinder (as is the case in the present diagram in which the clearances are neglected) then the latter, owing to its sudden expansion, experiences a sudden diminution of pressure or *drop*, which may cause a loss of effect shewn by the shaded area $EKXH$. But as, on the other hand, the L.P. initial pressure AP in the diagram is greater than the ultimate back-pressure in the H.P. cylinder, the H.P. and L.P. diagrams overlap each other in the cross-hatched surface $PFSX$ which, as it represents work done twice cover, first by the H.P. and then by the L.P. piston, is to be deducted from the area $EKXH$ representing the loss of effect. The residuary area even is not to be regarded as all loss, for by the mechanical theory of heat it is perfectly indifferent wheter the steam expands slowly or suddenly, its mechanical effect remains the same. In sudden expansion steam cannot exert all its work, so that part of it is converted into heat, i. e. the receiver steam is superheated to a certain extent, thus diminishing the condensation of the steam expanding after the drop, and its energy appears in the form of increased height of the L.P. expansion line, expressed by the shaded area HGU. The area enclosed between the expansion line and the MARIOTTE'S curve must theoretically be equal to the residual area $EKXH - PFSX$. But this is not the case in reality for there are always losses due partly to the friction of the steam in the passages and partly to the cooling in the receiver caused by convection and radiation. The advantage of a certain amount of drop is referred to again in § 52,26, which see.

Avoidance of losses due to sudden expansion.

16) The losses of effect due to sudden expansion, or more correctly to expansion without doing work, can be avoided by making $p_r'' = p\,\varepsilon$. We always have this in our power by so regulating the L.P. cut-off that the receiver space regarded merely *geometrically* becomes harmless. By Eq. 228.

$$p\,\varepsilon = p\,\varepsilon_g \left[\frac{\left(\frac{1}{\varepsilon_n} - 1\right) v}{R} + \frac{1}{\varepsilon_n} \right]$$

or

$$\frac{\varepsilon}{\varepsilon_g} = \frac{V}{v} = \frac{\left(\frac{1}{\varepsilon_n} - 1\right)v}{R} + \frac{1}{\varepsilon_n}$$

$$\frac{V}{v}\frac{R}{v} + 1 = \frac{1}{\varepsilon_n}\left(1 + \frac{R}{v}\right)$$

$$\varepsilon_n = \frac{\frac{R}{v} + 1}{\frac{V}{v}\frac{R}{v} + 1} \quad \ldots\ldots\ldots\ldots\ldots \quad (229)$$

If, in this expression $\frac{R}{v} = 0$ and $\varepsilon_n = 1$ it represents the old Woolf engine. For $\frac{R}{v} = 1$, that is the volume of the receiver equal to that of the H.P. cylinder, we get

$$\varepsilon_n = \frac{2}{\frac{V}{v} + 1}$$

and if further, $\frac{V}{v} > 3$, then $\varepsilon_n < {}^1/_2$, and a special valve-gear is required for the L.P. which will cut-off at less than half-stroke in order to make complete use of the expansion work of the steam.

17) The following table calculated by Eq. 229 gives the L.P. cut-offs for receiver compounds which will prevent the losses of effect mentioned in 15), due to the drop. Most favourable cut-off in L.P.

Table of L.P. cut-off ratios for Receiver Compounds.

Cylinder-ratio $\frac{V}{v}$		1	2	3	4	5	6
Cut-off ratio ε_n of	$\frac{R}{v} = 1$	1.00	0.66	0.50	0.40	0.33	0.28
" " " "	$\frac{R}{v} = 2$	1.00	0.60	0.43	0.33	0.27	0.23
" " " "	$\frac{R}{v} = 3$	1.00	0.57	0.40	0.30	0.25	0.21

18) If the losses of effect due to the cooling of the passing steam by the colder receiver walls are also to be avoided, i. e. if the receiver is to be rendered *calorically* harmless, it must be steam-jacketed, but this arrangement involves so much extra weight that it is only adopted in quite exceptional cases of marine practice. Avoidance of the losses of effect due to cooling.

§ 50.

Calculation for Two-cylinder Compounds.

Calculation of the diameter of L.P. cylinder.

1) **I. Determination of the cylinder diameters.** These engines, which are now rarely fitted to any but river steamers and are here always assumed to have their cranks at right-angles, are in general to be calculated exactly like WOOLF engines. As before, the first step is to determine the diameter of L.P. cylinder as if it did the whole *IHP* at the most favourable total cut-off ε_{g_v}, selected from the data in § 49, 2 according to the initial pressure p or the working pressure p_x. The cylinder-ratio $\frac{V}{v}$, the receiver volume R, and the L.P. cut-off ratio ε_n have no influence upon the *IHP*, but they affect the distribution of the power, so that several important qualities of the engine depend upon them (see § 28, 44). This dependence can again be most easily demonstrated by help of indicator diagrams. These diagrams having been constructed according to § 18, 39 to 46, the diagrams of twisting moments can easily be obtained from them as required for investigating the degree of uniformity (see § 48, and § 49, 5).

Empirical values for the reduced mean pressure p_{o_r}

2) After settling upon an absolute boiler pressure and a cylinder ratio $\frac{V}{v}$, the diameter D_n of the L.P. cylinder can be calculated by means of the following data as to the actual reduced mean pressures observed in war-ships and merchant-ships (§ 29, 15). For the same boiler pressure war-ships mostly have smaller cylinder ratios than merchant-ships (the reason of this is given in § 52, 5) and therefore rather higher reduced mean pressures. It is found approximately that

for naval engines with 5 atmos. working pressure $p_{o_r} = 1.8$ to 1.9 atmos.

for naval engines with 7 atmos. working pressure $p_{o_r} = 2.0$ to 2.2 atmos.

for mercantile engines with 5 to 7 atmos. working pressure $p_{o_r} = 1.4$ to 1.5 atmos.

for mercantile engines up to 10 atmos. working pressure $p_{o_r} = 2.2$ to 2.4 atmos.

Designing combined diagrams of two-cylinder compounds.

3) **II. Construction of the theoretical indicator diagrams.** If it is desired to shorten the draughtsman's work upon the first approximation, the following method of RANKINE'S can be used for rapidly sketching the combined diagram of a proposed two-cylinder compound, the losses of effect being neglected for the present. The circumscribed diagram $ACDGB$ having been drawn (Fig.

15, Pl. 11), the back pressure line MN is put in. Thereupon AL is made equal to the L.P. cut-off volume, so that $\frac{AL}{AB} = \varepsilon_n$. Draw LK perpendicular to AB, then K is the point of L.P. cut-off and at the same time the end of the curve PK separating the H.P. and L.P. diagrams, the construction of which curve is the chief object in view. If there are no steam-losses, the H.P. back-pressure curve, and the L.P. forward pressure curve coincide, or in other words PK is a pressure curve exhibiting the expansion of the receiver steam during the one part of the stroke and the compression of it during the other part. Regarding the curve in this light it is only necessary to make the length of AT such that it shall bear the same ratio to AB as the receiver volume does to the volume of the L.P. cylinder and in T to erect a perpendicular on which $TS = KL\frac{TL}{TA}$ is to be measured off. On drawing a parallel from S to TB, we obtain the starting point P of the boundary curve PK, which is a MARIOTTE'S curve whose pole lies in T, and is drawn according to § 6, 16. The H.P. diagram $CDKP$ and the L.P. diagram $PKGNM$ are thus determined.

4) In practice the admission of the steam *to* the receiver and its exit *from* it are only partially simultaneous, so that PK no longer shews the compression of the receiver steam and at the same time its fall of pressure at the L.P. admission. If besides, the steam-losses are regarded which arise from friction in the passages, convection, and radiation, the result is, as COWPER*) has demonstrated by actual cards, that the boundary curve PK of the L.P. diagram almost coincides with a straight line Kr parallel to AB, so that it is sufficiently accurate to regard Kr as the line of separation of the two diagrams. In the engines investigated by COWPER, the volume of the H.P. cylinder was rather smaller than the volume of steam per stroke admitted to the L.P., hence the former volume was represented by AF instead of AL, on which account also the H.P. terminal pressure EF was greater than the receiver pressure KL and a drop occurred. Between the H.P. diagram $CDEqr$ and the L.P. diagram $rKGNM$ there always remained therefore a kind of indent EKq representing the loss of effect, the reasons of which were explained at the conclusion of the preceding paragraph. For a more exact construction of the separate theoretical indicator diagrams for two-cylinder compounds the cal- Correction of the approximate diagram.

*) Transactions of the institution of naval architects 1864. p. 251 & Figs. 12 & 13, Pl. XIX.

Table of actual reduced mean pressures in two-cylinder Compound-Engines.

	Naval Engines									Mercantile Engines									
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Ship's name	"Charlotte"	"Alexandrine"	"Marie"	"Adler"	"Greif"	"Brummer"	"Schwalbe"	"Eber"	"Schütze"	"Borussia"	"Strassburg"	"Lotharingia"	"Hohenstaufen"	"Graf Bismarck"	"Bavaria"	"Rugia"	"Hungaria"	"Rovuma"	Remarks
Working pressure in atmos.	5	5	5	5	7	7	7	7	9	4.22	4.22	4.22	4.22	5.27	4.92	5.62	6.25	10	The pressures marked with a * were taken on voyages, all others on trial-trips.
Cylinder-ratio $\frac{V}{v}$	2.95	2.87	2.96	2.79	2.88	2.94	2.95	3.05	2.66	4.00	2.45	4.00	3.40	3.41	3.33	3.70	3.57	3.60	
Actual reduced mean pressure p_{o_r} in atmos.	1.88	1.78	1.85	1.85	2.17	2.27	2.41	2.20	3.72	1.55*	1.14	1.20*	1.02*	1.42	1.60	1.61*	1.46*	2.46	

culation of the principal pressures is indispensable. It is necessary to distinguish whether the L.P. cut-off occurs

a) after half stroke, or

b) before half stroke,

as this circumstance materially influences the distribution of steam in the engine. SCHRÖTER'S graphic method of determining the steam volumes (§ 51, 28) is preferable to numerical calculation as it shortens the work.

5) **a. L.P. cut-off ratio greater than 0.5.** Assuming an infinite connecting rod and the clearances to be neglected, the following general relations exist between the positions of piston and crank for each engine when the H.P. is the leading crank (Fig. 5, Pl. 15), β being the variable crank angle. **Positions of Cranks.**

I) $\measuredangle\, C\,O\,A = \beta < 90^0$ for the first half of stroke in the H.P.	II) $\measuredangle\, C\,O\,A_1 = \beta > 90^0$ for the second half of stroke in the H.P.

$O\,B$ . . position of H.P. crank . . $O\,A$

$O\,A$ „ „ L.P. „ $O\,A_1$

a) For the H.P. crank

$\frac{C\,F}{D\,C} = \frac{1 - \sin\beta}{2}$	$\frac{D\,E}{D\,C} = \frac{1 + \sin\beta}{2}$

b) For the L.P. crank

$\frac{D\,E}{D\,C} = \frac{1 + \cos\beta}{2}$	$\frac{D\,E_1}{D\,C} = \frac{1 - \cos\beta}{2}$

If the L.P. crank leads, or what is the same thing, when the present engine is going astern, analogous expressions are obtained which will not be discussed as they would lead too far, considering the comparatively small importance now-a-days of the compound engine.

6) Assuming the L.P. cut-off to take place at the position $O\,A$ of the L.P. crank, corresponding to the position $O\,B$ of the H.P. crank at the angle which we will call $\beta_1 = C\,O\,A$ we get **Contents of the H.P. cylinder at the moment of L.P. cut-off.**

$$\frac{D\,E}{D\,C} = \frac{1 + \cos\beta_1}{2} = \varepsilon_n$$

whence $$\cos\beta_1 = 2\,\varepsilon_n - 1$$

and $$\sin\beta_1 = \sqrt{1 - \cos^2\beta_1}$$

$$\sin\beta_1 = \sqrt{1 - (2\,\varepsilon_n - 1)^2}$$

$$\sin\beta_1 = 2\,\varepsilon_n \sqrt{\frac{1}{\varepsilon_n} - 1},$$

so that $$\frac{C\,F}{D\,C} = \frac{1 - 2\,\varepsilon_n \sqrt{\frac{1}{\varepsilon_n} - 1}}{2} = m,$$

as this expression shall be called for the sake of brevity. Consequently the volume of steam present in the H.P. cylinder and in communication with the L.P. just before cut-off in the latter is

$$v(1-m) = v\left(\frac{1 + 2\,\varepsilon_n \sqrt{\frac{1}{\varepsilon_n} - 1}}{2}\right)$$

The following table shews some values of $1 - m$ for varying ε_n

$\varepsilon_n = 0.50$	0.55	0.60	0.65	0.70	0.75	0.80
$1 - m = 1.000$	0.998	0.990	0.977	0.958	0.933	0.900

Terminal pressure in H.P. cylinder.

7) Having sketched the ideal diagram $ACDGB$ (Fig. 3, Pl. 15) as before described (§ 49, 6) and drawn in the back-pressure line so that $AM = a$, the point of the H.P. release E, is fixed by making $\frac{AL}{AB} = \frac{v}{V}$. The pressure at this place is

$$LE = p\,\varepsilon = \frac{p\,\varepsilon_g\,V}{v}$$

As shewn in § 49, 10, the steam exhausting from the H.P. cylinder suddenly falls in pressure from LE to LK on entering the receiver, causing the receiver pressure to rise suddenly from QH to QU, as further explained below. Thereupon the steam enclosed between the two pistons expands in consequence of the forward motion of the L.P. piston and the pressure sinks from $LK = QU$ to $VJ = ZW$.

Pressure in the L.P. at the cut-off.

8) As the steam in the L.P. cylinder expands $\frac{1}{\varepsilon_n}$ times, and its pressure at the end of the stroke is $p\,\varepsilon_g$, its pressure at cut-off must be

$$ZW = VJ = \frac{p\,\varepsilon_g}{\varepsilon_n}$$

The volume corresponding to this pressure and remaining in the receiver and in the H.P. cylinder in front of the piston after the L.P. slide has cut-off, is

$$R + v(1-m)$$

Receiver pressure at the beginning of the stroke in the L.P. p'_r

9) This steam is compressed by the H.P. piston until the beginning of the next stroke of the L.P. engine; its volume is then reduced to $R + \frac{v}{2}$ and its pressure raised to

$$\left[R + v(1-m)\right]\frac{p\,\varepsilon_g}{\varepsilon_n} = \left(R + \frac{v}{2}\right)p'_r$$

$$p'_r = \frac{\left[R + v\,(1-m)\right]\frac{p\,\varepsilon_g}{\varepsilon_n}}{R + \frac{v}{2}}$$

$$p'_r = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + (1-m)}{\frac{R}{v} + \frac{1}{2}}\right] \ldots\ldots\ldots\ldots (230)$$

i. e. the L.P. initial pressure is $AP = TS = BR$.

10) Consequently the ordinates of the curves JS and WR are calculated by substituting instead of $\frac{v}{2}$, the volume corresponding to each particular position of the crank, Receiver pressure during the L.P. expansion p'_n

$$v - v\left(\frac{1-\sin\beta}{2}\right) = v\left(1 - \frac{1-\sin\beta}{2}\right) = v\left(\frac{1+\sin\beta}{2}\right),$$

so that the ordinate for any point of the curve is

$$p'_n = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + (1-m)}{\frac{R}{v} + \left(\frac{1+\sin\beta}{2}\right)}\right] \ldots\ldots\ldots\ldots (231)$$

11) The steam at pressure $p'_r = AP$ enters the L.P. cylinder, drives the piston to half stroke and then has the volume Receiver pressure immediately before the H.P. release p''_r

$$R + \frac{V}{2},$$

because the H.P. piston is now at the end of its stroke; the pressure therefore changes to $QH = p''_r$.

$$\left[R + v\,(1-m)\right]\frac{p\,\varepsilon_g}{\varepsilon_n} = \left(R + \frac{V}{2}\right)p''_r$$

$$p''_r = \frac{\left[R + v\,(1-m)\right]\frac{p\,\varepsilon_g}{\varepsilon_n}}{R + \frac{V}{2}}$$

$$p''_r = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + (1-m)}{\frac{R}{v} + \frac{1}{2}\frac{V}{v}}\right] \ldots\ldots\ldots\ldots (232)$$

As during the L.P. admission, the driving pressure in this engine is the back-pressure in the H.P., we must have

$$p''_r = QH = AF,$$

the terminal back-pressure in the H.P.

12) The ordinates p''_n of the curves PH and SF are found by Eq. 231 if, instead of $R + \frac{V}{2}$, the volumes corresponding to Receiver pressure during the L.P. admission p''_n

the various crank positions

$$R+v\left(\frac{1-\sin\beta}{2}\right)+V\left(\frac{1+\cos\beta}{2}\right)\quad \text{are substituted,}$$

thus
$$p''_n=\frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v}+(1+m)}{\frac{R}{v}+\left(\frac{1-\sin\beta}{2}\right)+\frac{V}{v}\left(\frac{1+\cos\beta}{2}\right)}\right]\quad\ldots\ldots(233)$$

Receiver pressure immediately after the H.P. exhaust p_r

13) When the H.P. piston is on the centre and the L.P. at half stroke, the H.P. exhausts at pressure $p\,\varepsilon=\frac{p\,\varepsilon_g\,V}{v}$ into the receiver, the pressure in which was p''_r just before, but is now raised from $AF=QH$ to $LK-QV=p_r$ which is obtained from the relation

$$\frac{p\,\varepsilon_g}{\varepsilon_n}\Big[R+v\,(1-m)\Big]+v\,\frac{p\,\varepsilon_g\,V}{v}=p_r\left[v+R+\frac{V}{2}\right]$$

$$p_r=\frac{\frac{p\,\varepsilon_g}{\varepsilon_n}\Big[R+v\,(1-m)\Big]+v\,\frac{p\,\varepsilon_g\,V}{v}}{v+R+\frac{V}{2}}$$

$$p_r=\frac{p\,\varepsilon_g\left[\frac{R}{\varepsilon_n}+\frac{v}{\varepsilon_n}(1-m)+V\right]}{v+R+\frac{R}{2}}$$

$$p_r=\frac{p\,\varepsilon_g\left[\frac{1}{\varepsilon_n}\left\{\frac{R}{v}+(1-m)\right\}+\frac{V}{v}\right]}{1+\frac{R}{v}+\frac{1}{2}\frac{V}{v}}\quad\ldots\ldots\ldots\ldots(234)$$

Receiver pressure immediately before the L.P. cut-off p_n

14) There still remain the short pieces KJ and UW of the curve to be determined, the ordinates of which p_n are found from

$$p\,\frac{\varepsilon_g}{\varepsilon_n}\Big[R+v\,(1-m)\Big]+v\,\frac{p\,\varepsilon_g\,V}{v}=p_n\left[R+v\left(\frac{1+\sin\beta}{2}\right)+V\left(\frac{1-\cos\beta}{2}\right)\right]$$

$$p_n=\frac{p\,\varepsilon_g\left[\frac{1}{\varepsilon_n}\left\{\frac{R}{v}+(1-m)\right\}+\frac{V}{v}\right]}{\frac{R}{v}+\left(\frac{1+\sin\beta}{2}\right)+\frac{R}{v}\left(\frac{1-\cos\beta}{2}\right)}\quad\ldots\ldots\ldots(235)$$

Data.

15) All the curves are now computed and the diagram can be completely plotted. The data for that shewn in Fig. 3, Pl. 15 are

p = 4.66 atmospheres = absolute initial pressure,

a = 0.20 „ = back-pressure in condenser,

$\frac{V}{v}$ = 4.00 = cylinder-ratio,

$\frac{R}{v}$ = 1.00 = receiver-ratio,

$\varepsilon_g = 0.125 =$ ratio of cut-off, total,
$\varepsilon = 0.50 =$ „ „ in H.P.,
$\varepsilon_n = 0.55 =$ „ „ „ L.P.

16) It may be seen from the diagram that the steam on leaving the H.P. cylinder suddenly falls from the pressure LE to LK. In reality this drop EK is not so great, because the steam is to some extent wire-drawn in the passages and thus only reaches the receiver gradually; EKJ is therefore somewhat convex to AB. What was said in § 49, 15 as to the shaded areas representing losses of effect also applies here. **Drop.**

17) b. **Cut-off in L.P. less than 0.5.** The variable crank angle being again (Fig. 6, Pl. 15) called β, **Positions of Cranks.**

I) $\sphericalangle DOA = \beta < 90^0$ corresponding to second half of stroke in H.P.	II) $\sphericalangle DOB = \beta < 90^0$ corresponding to first half of stroke in H.P.

OB . . position of H.P. crank . . OB_1
OA „ „ L.P. „ OB,

the relations worked out in 5) take the following form

a) For the H.P. crank,

$$\frac{DF}{DC} = \frac{1 + \sin\beta}{2} \qquad\qquad \frac{CF}{DC} = \frac{1 - \sin\beta}{2}$$

b) For the L.P. crank,

$$\frac{DE}{DC} = \frac{1 - \cos\beta}{2} \qquad\qquad \frac{DF}{DC} = \frac{1 + \cos\beta}{2}.$$

Calling the L.P. crank angle β_1 for the position OA at L.P. cut-off, we get

$$\frac{DE}{DC} = \beta_n = \frac{1 - \cos\beta_1}{2}$$

$$\frac{DF}{DC} = m = \frac{1 + \sin\beta_1}{2}$$

$$\cos\beta_1 = 1 - 2\,\varepsilon_n$$

$$\sin\beta_1 = \sqrt{1 - \cos^2\beta} = \sqrt{1 - (1 - 2\,\varepsilon_n)^2}$$

$$\sin\beta_1 = 2\,\varepsilon_n \sqrt{\frac{1}{\varepsilon_n} - 1}$$

$$m = \frac{1 + 2\,\varepsilon_n \sqrt{\frac{1}{\varepsilon_n} - 1}}{2}$$

$$1 - m = \frac{1 - 2\varepsilon_n \sqrt{\frac{1}{\varepsilon_n} - 1}}{2}.$$

The following table contains some values of $1 - m$ for varying cut-off ratios ε_n.

ε_n	0.20	0.25	0.30	0.33	0.40	0.45
$1-m$	0.1000	0.0670	0.0417	0.0226	0.0111	0.0025

Pressure in L.P. cylinder at cut-off.

18) The ideal diagram $ACDGB$, Fig. 4, Pl. 15, having been sketched, the point E and the ordinate LE found as explained in 7), the various portions of the boundary curve between the two diagrams are determined as follows. The pressure in the receiver at L.P. cut-off is

$$QH = \frac{p\,\varepsilon_g}{\varepsilon_n} = VJ$$

and the volume of the steam left in it

$$R + v\,(1-m).$$

Receiver pressure immediately before the H.P. exhaust. p_r''

19) At L.P. half-stroke this volume is diminished to the contents R of the receiver, the pressure in the latter is therefore raised from $QH = VJ$ to $ZU = AF = p_r''$ and is calculated thus,

$$\frac{p\,\varepsilon_g}{\varepsilon_n}\left[R + v\,(1-m)\right] = p_r''\,R$$

$$p_r'' = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{R + v\,(1-m)}{R}\right]$$

$$p_r'' = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + (1-m)}{\frac{R}{v}}\right] \quad \ldots\ldots\ldots\ldots (236)$$

Receiver pressure immediately after L P. cut-off. p_n^0

20) From this formula the ordinates of the portions JF and HU of the curve are obtained by substituting for R in the denominator of the expression in the bracket the volume

$$R + \left(\frac{1-\sin\beta}{2}\right)v,$$

corresponding to the position of the crank. The value of any ordinate p_n^0 is therefore

$$p_n^0 = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + (1-m)}{\frac{R}{v} + \frac{1-\sin\beta}{2}}\right] \quad \ldots\ldots\ldots\ldots (237)$$

Receiver pressure immediately after the H.P. exhaust p_r

21) At L.P. half stroke the H.P. cylinder discharges its volume v at pressure $p\,\varepsilon = \frac{p\,\varepsilon_g V}{v} = LE$ into the receiver and the pressure ZU in this rises to $ZW = LK = p_r$, which is calculated as follows

$$\frac{p\,\varepsilon_g}{\varepsilon_n}\left[R + v\,(1-m)\right] + \frac{p\,\varepsilon_g\,V}{v}\,v = (v+B)\,p_r$$

$$p_r = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{R + v\,(1-m) + V\varepsilon_n}{v+R}\right]$$

$$p_r = p\,\varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left(\frac{R}{v} + (1-m) + \frac{V}{v}\right)}{1 + \frac{R}{v}}\right] \quad \ldots\ldots (238)$$

22) At the end of the L.P. stroke the receiver steam assumes the volume $R + \frac{v}{2}$ to which it has been compressed, its pressure has therefore risen from $ZW = LK$ to $BR = TS = AP = p'_r$, the L.P. initial pressure, which is calculated from Receiver pressure at beginning of L.P. stroke p'_r

$$p\,\varepsilon_g \left[\frac{R}{\varepsilon_n} + \frac{v}{\varepsilon_n}(1-m) + V\right] = \left(R + \frac{v}{2}\right) p'_r$$

$$p'_r = p\,\varepsilon_g \left[\frac{\frac{R}{\varepsilon_n} + \frac{v}{\varepsilon_n}(1-m) + V}{R + \frac{v}{2}}\right]$$

$$p'_r = p\,\varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left[\frac{R}{v} + (1-m)\right] + \frac{V}{v}}{\frac{R}{v} + \frac{1}{2}}\right] \quad \ldots\ldots (239)$$

23) The ordinates p'_n of the curves KS and WR are obtained from the first of the three preceding formulæ by inserting Receiver pressure during the L.P. expansion p'_n

$$R + v\left(\frac{1 + \sin\beta}{2}\right)$$

instead of $R + \frac{v}{2}$, which gives

$$p_n = p\,\varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left\{\frac{R}{v} + (1-m)\right\} + \frac{V}{v}}{\frac{R}{v} + \frac{(1 + \sin\beta)}{2}}\right] \quad \ldots\ldots. (240)$$

24) To complete the diagram the curves PH and SJ are still wanting, their ordinates p''_n are found from Receiver pressure during L.P. admission p''_n

$$\frac{p\,\varepsilon_g}{\varepsilon_n}\left[R + v(1-m) + V\varepsilon_n\right] = p''_n\left[R + v\left(\frac{1-\sin\beta}{2}\right) + V\left(\frac{1+\cos\beta}{2}\right)\right]$$

$$p''_n = p\,\varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left\{\frac{R}{v} + (1-m)\right\} + \frac{V}{v}}{\frac{R}{v} + \frac{1-\sin\beta}{2} + \frac{V}{v}\left(\frac{1+\cos\beta}{2}\right)}\right] \quad \ldots\ldots (241)$$

25) The diagram can now be drawn; that shewn by Fig. 4, Pl. 15 is adapted to the data in 15) with one difference, viz that $\varepsilon_n = 0.4$ instead of 0.55. This figure shews that in this case also the passage of the steam from the H.P. cylinder to the receiver is accompanied by a drop in the former and a rise of Drop.

pressure in the latter. But this drop is neither so great nor so injurious as when $\varepsilon_n > 0.5$, as it only increases the pressure in the L.P. at the beginning of the stroke and does not cause it to rise in the middle of the stroke. As however the shaded surfaces shew, the whole expansion work of the steam is here again incompletely utilized, losses of effect still being present.

Formulae collated. 26) For greater perspicuity the leading particulars of the diagrams Figs. 3 and 4, Pl. 15 and the quantities and formulæ for their computation are collated below.

α) H.P. volume, $AL = v$,

β) L.P. " $AB = V$,

γ) total ratio of cut-off $\frac{CD}{AB} = \varepsilon_g$,

δ) L.P. " " "

$\varepsilon_n > 0{,}5$; $\frac{AZ}{AB} = \varepsilon_n$, $\qquad$ $\varepsilon_n < 0{,}5$; $\frac{AQ}{AB} = \varepsilon_n$,

ε) H.P. initial pressure,

$AC = p$ in atmos. absolute,

ζ) back pressure in condenser,

$AM = BN = a$,

η) L.P. terminal pressure,

$BG = p\,\varepsilon_g$,

ϑ) Curve DEG, MARIOTTE'S curve, constructed by § 49,6 or § 6,16.

ι) H.P. terminal pressure,

$$LE = p\,\varepsilon = \frac{p\,\varepsilon_g\,V}{v},$$

κ) receiver-pressure immediately after H.P. exhaust, i. e. at L.P. half-stroke,

$$LK = QU = p_r = p\,\varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left\{\frac{R}{v} + (1-m)\right\} + \frac{V}{v}}{1 + \frac{R}{v} + \frac{1}{2}\frac{V}{v}}\right] \text{ for } \varepsilon_n > 0{,}5$$

$$LK = ZW = p_r = p\,\varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left\{\frac{R}{v} + (1-m)\right\} + \frac{V}{v}}{1 + \frac{R}{v}}\right] \text{ for } \varepsilon_n < 0{,}5,$$

λ) ordinate of the curve

KJ or UW for $\varepsilon_n > 0{,}5$ $\qquad$ JF or HU for $\varepsilon_n < 0{,}5$

$$p_n = p\,\varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left\{\frac{R}{v} + (1-m)\right\} + \frac{V}{v}}{\frac{R}{v} + \frac{1+\sin\beta}{2} + \frac{V}{v}\left(\frac{1-\cos\beta}{2}\right)}\right] \qquad p_n^0 = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + (1-m)}{\frac{R}{v} + \frac{1-\sin\beta}{2}}\right]$$

μ) receiver pressure immediately before L.P. cut-off

$$VJ = ZW = \frac{p\,\varepsilon_g}{\varepsilon_n};\ \text{for } \varepsilon_n > 0{,}5 \qquad VJ = QH = \frac{p\,\varepsilon_g}{\varepsilon_n};\ \text{for } \varepsilon_n < 0{,}5,$$

ν) Ordinate of the curve

JS or WR for $\varepsilon_n > 0{,}5$ — KS or WR for $\varepsilon_n < 0{,}5$

$$p'_n = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + (1-m)}{\frac{R}{v} + \frac{1+\sin\beta}{2}}\right] \qquad p'_n = p\,\varepsilon_g\left[\frac{\frac{1}{\varepsilon_n}\left\{\frac{R}{v} + (1-m)\right\} + \frac{V}{v}}{\frac{R}{v} + \frac{1+\sin\beta}{2}}\right],$$

ξ) receiver pressure when L.P. engine is on the centre, i. e. L.P. initial pressure,

$$TS = AP = BR = p'_r = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + (1-m)}{\frac{R}{v} + \frac{1}{2}}\right] \text{ for } \varepsilon_n > 0{,}5$$

$$TS = AP = BR = p'_r = p\,\varepsilon_g\left[\frac{\frac{1}{\varepsilon_n}\left\{\frac{R}{v} + (1-m)\right\} + \frac{V}{v}}{\frac{R}{v} + \frac{1}{2}}\right] \text{ for } \varepsilon_n < 0{,}5,$$

o) Ordinate of the curve,

SF or PH

$$p''_n = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + (1-m)}{\frac{R}{v} + \frac{1-\sin\beta}{2} + \frac{V}{v}\left(\frac{1+\cos\beta}{2}\right)}\right] \text{ for } \varepsilon_n > 0{,}5,$$

SJ or PH

$$p''_n = p\,\varepsilon_g\left[\frac{\frac{1}{\varepsilon_n}\left\{\frac{R}{v} + (1-m)\right\} + \frac{V}{v}}{\frac{R}{v} + \frac{1-\sin\beta}{2} + \frac{V}{v}\left(\frac{1+\cos\beta}{2}\right)}\right] \text{ for } \varepsilon_n < 0{,}5,$$

π) receiver pressure immediately before H.P. release,

$$AF = QH = p''_r = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + (1-m)}{\frac{R}{v} + \frac{1}{2}\frac{V}{v}}\right] \text{ for } \varepsilon_n > 0{,}5,$$

$$AF = ZU = p''_r = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + (1-m)}{\frac{R}{v}}\right] \text{ for } \varepsilon_n < 0.5,$$

27) As already mentioned in 1) the various characteristics of a compound are dependent upon the choice of a cylinder-ratio $\frac{V}{v}$ and of a cut-off ratio ε_n for the L.P. The receiver-ratio $\frac{R}{v}$ is **Determination of receiver-volume.**

absolutely indifferent. In practice the receiver volume is usually made equal to the volume of the H.P. cylinder, that is $\frac{R}{v} = 1$, it is seldom larger in marine engines on account of the weight.

Various points of view in designing compounds.

28) **III. Determining the cylinder-ratio.** The points of view from which the choice of a cylinder-ratio for a compound may be regarded are briefly the following,

a) the drop described in 16) and 25) is to be avoided,

b) both pistons are to do equal work,

c) the maximum stresses on the rods are to be as low as possible,

d) the twisting moment is to be as uniform as possible.

Avoidance of Drop.

29) a) **The drop,** so far as it is due to circumstances of volume, can be avoided if (Figs. 3 and 4, Pl. 15), we make

$$LE = LK,$$

we then have for

$\varepsilon_n > 0{,}5$

$$\frac{p\,\varepsilon_g V}{v} = p\varepsilon_g\left[\frac{\frac{R}{\varepsilon_n}+\frac{v}{\varepsilon_n}(1-m)+V}{v+R+\frac{V}{2}}\right]$$

$$\frac{V}{v} = \frac{\frac{R}{\varepsilon_n}+\frac{v}{\varepsilon_n}(1-m)+V}{v+R+\frac{V}{2}}$$

$$\frac{v}{\varepsilon_n V} = \frac{v+R+\frac{V}{2}}{R+v(1-m)+\varepsilon_n V}$$

$$\frac{v}{\varepsilon_n V} = \frac{1+\frac{R}{v}+\frac{V}{2v}}{(1-m)+\frac{R}{v}+\frac{\varepsilon_n V}{v}}$$

$$\frac{1}{\varepsilon_n}\left[(1+m)+\frac{R}{v}\right]+\frac{V}{v} = \frac{V}{v}+\frac{V}{v}\left(\frac{R}{v}+\frac{V}{2v}\right)$$

$$\frac{1}{\varepsilon_n}\left[(1-m)+\frac{R}{v}\right] = \frac{V}{v}\frac{R}{v}+\frac{1}{2}\left(\frac{V}{v}\right)^2$$

$$\frac{V}{v} = \sqrt{\left(\frac{R}{v}\right)^2+\frac{2}{\varepsilon_n}\left[(1-m)+\frac{R}{v}\right]} = \frac{R}{v} \quad (242)$$

$\varepsilon_n < 0{,}5$

$$\frac{p\,\varepsilon_g V}{v} = p\,\varepsilon_g\left[\frac{\frac{1}{\varepsilon_n}\left[R+v(1-m)\right]+V}{v+R}\right]$$

$$\frac{V\varepsilon_n}{v}\,\frac{(1-m+\frac{R}{v}+\frac{V\varepsilon_n}{v}}{1+\frac{R}{v}}$$

$$\frac{V}{v}\left(1+\frac{R}{v}\right) = \frac{1}{\varepsilon_n}\left[(1-m)+\frac{R}{v}\right]+\frac{V}{v}$$

$$\frac{V}{v}\left(R+\frac{R}{v}\right)-\frac{V}{v} = \frac{1}{\varepsilon_n}\left[(1-m)+\frac{R}{v}\right]$$

$$\frac{V}{v}\left[1+\frac{R}{v}-1\right] = \frac{1}{\varepsilon_n}\left[(1-m)+\frac{R}{v}\right]$$

$$\frac{V}{v} = \frac{1}{\varepsilon_n}\left[\frac{(1-m)+\frac{R}{v}}{\frac{R}{v}}\right]. \quad (243)$$

If ε_n and the receiver-ratio $\frac{R}{v}$ are known, the proper cylinder-ratio can be determined from these formulæ. The following table was computed in that way.

Table of Cylinder-ratios of Two-cylinder Compounds in which no Drop occurs.

ε_n		0.20	0.25	0.30	0.35	0.40	0.45	0.50	0.55	0.60	0.65	0.70	0.75	0.80
$\frac{1}{\varepsilon_n}$		5.00	4.00	3.33	2.86	2.50	2.22	2.00	1.82	1.67	1.54	1.43	1.33	1.25
$\frac{V}{v}$ for	$\frac{R}{v} = 1$	5.50	4.27	3.48	2.92	2.52	2.23	2.00	1.88	1.76	1.66	1.57	1.48	1.40
	$\frac{R}{v} = 2$	5.25	4.13	3.41	2.89	2.51	2.22	2.00	1.86	1.74	1.63	1.53	1.44	1.36
	$\frac{R}{v} = 3$	5.17	4.09	3.38	2.88	2.50	2.22	2.00	1.85	1.73	1.61	1.51	1.42	1.33

30) From this table it may be seen that for a cylinder-ratio $\frac{V}{v} > 2$, the L.P. must cut-off *before* half stroke if no drop is to take place, also that $\frac{1}{\varepsilon_n}$ is about $= \frac{V}{v}$ and approaches it the more closely the larger the receiver is, consequently with an infinite receiver the L.P. cut-off ratio is simply the reciprocal of the cylinder ratio, i. e.

$$\varepsilon_n = \frac{v}{V} \quad \dots\dots\dots\dots\dots\dots \quad (244)$$

If this L.P. cut-off ratio which affords the most perfect utilizing of the steam is to be retained under any circumstances, a special expansion valve must be fitted to the L.P. engine in certain cases.

31) *The losses of effect* in the receiver which arise from a different value of ε_n can be calculated by the above formulæ. A result sufficiently accurate for practice is however obtained from RANKINE'S combined diagram Fig. 15, Pl. 11, by regarding the shaded area EKq as loss of effect and EK as a straight line; then

Calculation of the losses of effect due to circumstances of volume.

$$\text{triangle } EKq = \frac{1}{2}\left(\frac{p\,\varepsilon_g\,V}{v} - \frac{p\,\varepsilon_g}{\varepsilon_n}\right)\left(\varepsilon_n\,V - v\right), \text{ and}$$

$$\text{triangle } EKq = \frac{1}{2}p\,\varepsilon_g\,V\left(\frac{V}{v} - \frac{1}{\varepsilon_n}\right)\left(\varepsilon_n - \frac{v}{V}\right).$$

The total energy represented by the ideal diagram (see Eq. 111) is

$$p\,\varepsilon_g\,V\left(1 + \log nat \frac{1}{\varepsilon_g}\right);$$

The ratio of the loss to the energy actually produced is therefore

$$\frac{\left(\frac{V}{v} - \frac{1}{\varepsilon_n}\right)\left(\varepsilon_n - \frac{v}{V}\right)}{2\left(1 + \log\ nat \frac{1}{\varepsilon_g}\right)} \quad \dots\dots\dots\dots \quad (245)$$

The losses in the table on p. 464 are calculated by this formula. This table shews that losses due to the drop caused by certain proportions of the *volumes* may exceed 25°/₀, but do not exceed 5°/₀ for the values of ε_n usually occurring in practice. The losses may be greater in reality as the *caloric* influence of the receiver walls, so far not regarded, can only be obviated by jacketing the receiver, which is almost unknown in marine practice (Compare § 49, 18). But even supposing the other unfavourable influences to be removed, there remain the losses due to the friction of the steam in the passages. The particular circumstances under which a drop can occur without being accompanied by loss are explained in § 52,26.

Equal work on both pistons.

32) b. **Equal distribution of work on both pistons is attainable,** when

α) ε_n is given, — by a suitable choice of cylinder ratio $\frac{V}{v}$ and

β) $\frac{V}{v}$ " " — " " " " " ε_n.

Determination of a cylinder-ratio.

33) α. *The L.P. cut-off ratio ε_n being given,* the work done in the H.P. must be put equal to half the total *IHP*. The latter is by 31)

$$p\,\varepsilon_g\frac{V}{2}\left(1+\log\ nat\frac{1}{\varepsilon_g}\right).$$

If the losses of effect in the receiver are to be avoided as far as possible, then the L.P. cut-off ratio ε_n must equal $\frac{V}{v}$ and we get tho H.P. back-pressure

$$\frac{p\,\varepsilon_g}{\varepsilon_n}=\frac{p\,\varepsilon_g V}{v}.$$

The H.P. cut-off ratio is

$$\varepsilon=\frac{\varepsilon_g V}{v}$$

and the work done in the H.P. is therefore

$$v\,p\left[\varepsilon_g\frac{V}{v}\left(1+\log\ nat\frac{v}{\varepsilon_g V}\right)-\varepsilon_g\frac{V}{v}\right].$$

But this must be equal to that mentioned above

$$p\,\varepsilon_g\frac{V}{2}\left(1+\log\ nat\ \frac{1}{\varepsilon_g}\right)=v\,p\left[\varepsilon_g\frac{V}{v}\left(1+\log\ nat\ \frac{v}{\varepsilon_g V}\right)-\varepsilon_g\frac{V}{v}\right]$$

$$\frac{1}{2}\frac{V}{v}\left(1+\log\ nat\ \frac{1}{\varepsilon_g}\right)=\frac{V}{v}\left(1+\log\ nat\ \frac{v}{\varepsilon_g V}\right)-\frac{V}{v}$$

$$\frac{1+\log\ nat\,\frac{1}{\varepsilon_g}}{2}=\left(1+\log\ nat\frac{v}{\varepsilon_g V}\right)-1$$

$$\log\ nat\frac{v}{\varepsilon_g V}=\log\ nat\left(\frac{1}{\varepsilon_g}\times\frac{v}{V}\right)=\log\ nat\left(\frac{1\,.\,V}{\varepsilon_g\,.\,v}\right)=\log\ nat\,\frac{1}{\varepsilon_g}-\log\ nat\,\frac{V}{v}$$

$$\frac{1 + \log nat \frac{1}{\varepsilon_g}}{2} = 1 + \log nat \frac{1}{\varepsilon_g} - \log nat \frac{V}{v} - 1$$

$$\log nat \frac{V}{v} = \log nat \frac{1}{\varepsilon_g} = \left(\frac{1 + \log nat \frac{1}{\varepsilon_g}}{2} \right)$$

$$\log nat \frac{V}{v} = \frac{\log nat \frac{1}{\varepsilon_g} - 1}{2} \quad \dots\dots\dots\dots (246)$$

If this formula is used for working out the cylinder-ratio, we are neglecting the back-pressure in the condenser and the losses in the receiver which will occur in spite of the favourable cut-off-ratio chosen, but if the back-pressure and the receiver losses are to be regarded, in order to get a more accurate result, the losses can be added to the back-pressure, then called a_1, and the exact formula is

$$\log nat \frac{V}{v} = \frac{\log nat \frac{1}{\varepsilon_g} - 1}{2} - \frac{a_1}{p\, \varepsilon_g} \quad \dots\dots\dots (247)$$

34) *β. In determining the cut-off ratio* ε_n with an assumed cylinder-ratio $\frac{V}{v}$, we have it in our power to make the former ratio such as to divide the total *IHP* between the two cylinders in any proportion we please, involving of course more or less loss of effect. It is difficult to decide which is the best distribution of work between the cylinders, for compounds having cut-offs which should lead to equal powers, never shew them in reality, but the L.P. usually indicates only about $^2/_3$ the power of the H.P. As a rule the distribution is made so as to fulfil the condition c) or d) (see 28), that is, a cylinder-ratio $\frac{V}{v}$ being assumed, ε_n is calculated so as to suit the above conditions. RANKINE says both cylinders will do about equal work if we make **Determination of the cut-off ratio ε_n**

$$\varepsilon_n = \sqrt{\varepsilon_g} \quad \dots\dots\dots\dots\dots\dots (248)$$

35) c. **The stress on the rods is a minimum** when the maximum or, which is the same thing, the initial loads in both engines are equal. Referring to Eq. 230, and equating these initial pressures, we get when $\varepsilon_n > 0.5$, **Minimum stresses in the rods.**

$$v\left(p - \frac{p\, \varepsilon_g}{\varepsilon_n}\right) = V\left\{\frac{p\, \varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + (1 - m)}{\frac{R}{v} + \frac{1}{2}}\right] - a_1\right\}, \quad \dots (249)$$

or by Eq. 239, when $\varepsilon_n < 0{,}5$,

$$v\left(p - \frac{p\,\varepsilon_g}{\varepsilon_n}\right) = V\left\{p\,\varepsilon_g\left[\frac{\frac{1}{\varepsilon_n}\left\{\frac{R}{v} + (1-m)\right\} + \frac{V}{v}}{\frac{R}{v} + \frac{1}{2}}\right] - a_1\right\} \quad (250)$$

When ε_n is equal to, or only differs slightly from 0.5, the following approximate formula can be used

$$v\left(p - \frac{p\,\varepsilon_g}{\varepsilon_n}\right) = V\left(\frac{p\,\varepsilon_g}{\varepsilon_n} - a_1\right) \quad (251)$$

where a_1 again expresses the losses of pressure due to imperfect vacuum, friction in the passages &c. In these formulæ all the quantities are known, so that ε_n can easily be calculated. Assuming $a_1 = \frac{p}{15}$ we find for

Cylinder-ratio $\frac{V}{v}$		4.00	3.00	2.00
for $\varepsilon_g = {}^1/_{12}$	ε_n	0.55	0.33	0.22
	$\frac{1}{\varepsilon_n}$	1.80	3.00	4.60
for $\varepsilon_g = {}^1/_6$	ε_n	0.66	0.55	0.50
	$\frac{1}{\varepsilon_n}$	1.50	1.80	2.00

Initial pressure.

36) From the table in 44) it is evident in the first place that the distribution of work in the two cylinders is almost independent of the cylinder-ratio $\frac{V}{v}$ and in the second place that for a given value of ε_n the work done in the H.P. remains nearly constant whether its volume is 0.5 or 0.25 of that of the L.P. But the H.P initial pressure $v\left(p - \frac{p\,\varepsilon_g}{\varepsilon_n}\right)$ varies for the same work directly as the volume v, so that the initial pressure becomes the smaller, the smaller v is taken, of course assuming v to be kept greater than $V\varepsilon_g$. In the single-expansion engine the volume of each cylinder $= \frac{V}{2}$ and the initial pressure $p\,\frac{V}{2}$, so that this pressure is always greater than the corresponding pressure in a compound, the cylinder ratio of which $\frac{V}{v} > 2$.

Uniformity of twisting moments.

37) d. **Uniformity of Twisting Moments** *) can be attained by a due distribution of work in the two cylinders, but no fixed ratio for this can be

*) J. E. Armengaud Ainé. Traité théorique et pratique des moteurs à vapeur. Paris 1863.

prescribed, because it varies with $\frac{V}{v}$ and the total cut-off ratio ε_g. The following considerations shew however that it is possible to evolve from known quantities a value for ε_n which nearly fulfils the required condition. The diagram of twisting moments, Fig. 16, Pl. 11, constructed according to § 49,5, but without regarding inertia, shews that in every stroke as well as in every revolution two maximum moments occur, one for the H.P., the other for the L.P. If the ordinates y_h and y_n representing these moments, were equal, the ratio of maximum to mean moment would be the smallest. As the ordinates of the combined diagram depend upon both cylinders, and the expressions which represent them are very complicated, it is usual to be satisfied with making the maximum H.P. ordinate equal to the maximum L.P. ordinate, i. e. the maximum H.P. twisting moment equal to the maximum L.P. twisting moment. It is true that this method only attains an approximate uniformity of the twisting moment.

38) The greatest L.P. twisting moment occurs in general at cut-off, i. e. either before or after half-stroke, according as the cut-off ratio is **Maximum L.P. twisting moment.**

α) less than 0.5, or

β) greater than 0.5.

The general expression for an L.P. twisting moment (assuming an infinite connecting rod), results from Fig. 13, Pl. 11 as

$$P\, r \sin \beta,$$

but the piston-load P is

$$\frac{\pi}{4} D_n^2 p_n''$$

and consequently the twisting moment

$$M_{t_n} = \frac{\pi}{4} D_n^2 p_n'' r \sin \beta$$

or, as

$$\frac{\pi}{4} D_n^2 r = \frac{V}{2}$$

$$M_{t_n} = \frac{V}{2} p_n'' \sin \beta \quad \ldots\ldots\ldots\ldots \quad (252)$$

From this general expression the above two special cases are to be derived.

39) *α*) *The L.P. cut-off ratio ε_n is assumed < 0.5.* Substituting in Eq. 252 the value of p_n'', we get **Cut-off ratio $\varepsilon_n < 0.5$**

$$M_{t_n} = \frac{V}{2} p\, \varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left\{\frac{R}{v} + (1 - m)\right\} + \frac{V}{v}}{\frac{R}{v} + \frac{1 - \sin\beta}{2} + \frac{V}{v}\left(\frac{1 + \cos\beta}{2}\right)} \right] \sin \beta$$

This twisting moment attains its greatest value when the function of β is a maximum. To determine this maximum the small quantity $(1 - m)$ is neglected and the function of β evolved from the above expression as follows

$$\frac{\sin\beta}{2R + v + V - v\sin\beta + V\cos\beta}$$

$$2R + v + V = C = const.$$

$$f(\beta) = \frac{\sin\beta}{C - v\sin\beta + V\cos\beta};$$

whence, differentiating and equating to zero, we get

$$df(\beta) = (C - v\sin\beta + V\cos\beta)\cos\beta + \sin\beta\,(v\cos\beta + V\sin\beta) = 0$$

$$C\cos\beta - v\sin\beta\cos\beta + V\cos^2\beta + v\sin\beta\cos\beta + V\sin^2\beta = 0$$

$$C\cos\beta + V(\cos^2\beta + \sin^2\beta) = 0$$

$$\cos\beta = -\frac{V}{C} = -\frac{V}{v + V + 2R}.$$

For this value of β, the function of β is a maximum. Substituting this value of β in the above expression and again neglecting $(1 - m)$, ε_n can be derived as shewn below by equating the expression thus transformed to the maximum H.P. twisting moment.

Equating the maximum twisting moments of both engines.

40) Taking back-pressure into account, the greatest H.P. twisting moment is by the preceding

$$\frac{v}{2}\left(p - \frac{p\,\varepsilon_g}{\varepsilon_n}\right)\sin\gamma,$$

where γ is the H.P. crank angle at H.P. cut-off. Analogously to 6) $\sin\gamma$ can be thus expressed in terms of ε the H.P. cut-off ratio,

$$\sin\gamma = 2\,\varepsilon\sqrt{\frac{1}{\varepsilon} - 1}$$

and accordingly, equating both maximum twisting moments, we get

$$\frac{v}{2}\left(p - \frac{p\,\varepsilon_g}{\varepsilon_n}\right)2\,\varepsilon\sqrt{\frac{1}{\varepsilon} - 1} = \frac{\frac{V}{2}p\,\varepsilon_g\left[\frac{R}{v\,\varepsilon_n} + \frac{V}{v}\right]\sin\beta}{\frac{R}{v} + \frac{1 - \sin\beta}{2} + \frac{V}{v}\left(\frac{1 + \cos\beta}{2}\right)} - V a_1 \sin\beta \quad (253)$$

Cut-off ratio $\varepsilon_n > 0.5$.

41) β. *The L.P. cut-off ratio ε_n being assumed > 0.5*, the maximum L.P. twisting moment occurs at about half-stroke and has the value

$$\frac{V}{2}\left(\frac{p\,\varepsilon_g}{\varepsilon_n} - a_1\right)\sin\beta$$

and ε_n is then to be calculated from

$$\frac{v}{2}\left(p - \frac{p\,\varepsilon_g}{\varepsilon_n}\right)2\,\varepsilon\sqrt{\frac{1}{\varepsilon} - 1} = \frac{V}{2}\left(\frac{p\,\varepsilon_g}{\varepsilon_n} - a_1\right)\sin\beta \;\ldots\ldots\; \mathbf{(254)}$$

42) If however the connecting rod is not regarded as infinite, the piston-load is not to be multiplied by sin β, but according to § 78,3, by **Cut-off ratio ε_n for given cylinder ratios.**

$$\frac{\sin(a+\beta)}{\cos a}$$

The ratios given in the table in 44) are calculated for an infinite connecting rod, the drop being neglected, and the back-pressure taken at $\frac{p}{30}$. Under these suppositions the preceding formulæ give for

Cylinder-ratio $\frac{V}{v}$		4.00	3.00	2.00
for $\varepsilon_g = {}^1/_{12}$	ε_n	0.40	0.31	0.26
	$\frac{1}{\varepsilon_n}$	2.50	3.19	3.90
for $\varepsilon_g = {}^1/_6$	ε_n	0.77	0.62	0.50
	$\frac{1}{\varepsilon_n}$	1.30	1.60	2.00

43) These values agree with those of the table in 44) which shews that the formulæ are sufficiently exact for practical purposes. The cut-off ratios ε_n derived from the formulæ do not differ greatly from those which will give the minimum stresses on the rods, but in certain cases they are accompanied by considerable losses due to drop. It is however generally speaking possible for any total cut-off ratio ε_g, to hit upon a cylinder-ratio which approximately fulfils all the four conditions referred to in 28); for instance if for $\varepsilon_g = {}^1/_{12}$, $\frac{V}{v}$ is made $= 3$, or for $\varepsilon_g = {}^1/_6$, $\frac{V}{v} = 2$. We may remark that for very high expansion and a low cylinder-ratio, the particular L.P. cut-off ratio ε_n which gives the most equable twisting moment and the least stresses on the rods causes a back-pressure in the receiver greater than the terminal forward pressure in the H.P.; for instance with $\frac{V}{v} = 2$ and $\varepsilon_g = {}^1/_{12}$. Besides, experience teaches that excessive expansion is incompatible with uniform twisting moments (compare § 45,22) because during a portion of the stroke one engine is pulling the other round. **Drawbacks of too high expansion.**

44) The following table*) contains various values of $\frac{V}{v}$ and ε_n with the resulting characteristics of the engine. **Table of proportions of two-cylinder compounds.**

*) Naval Science. 1873. Vol. II. On compound engines.

Table of Proportions of Two-cylinder Compounds.

Compound		Total cut-off ratio $\varepsilon_g = {}^1/_{12}$			Total cut-off ratio $\varepsilon_g = {}^1/_6$		
Cylinder-ratio $\frac{V}{v}$	L.P. cut-off ratio ε_n	Loss due to drop in % of total work	Ratio of H.P. to L.P. work	Ratio of max. to mean twisting moment	Loss due to drop in % of total work	Ratio of H.P. to L.P. work	Ratio of max. to mean twisting moment
1	2	3	4	5	6	7	8
4	1.00	22.6	4.00	1.74	26.0	1.70	1.24
4	0.66	14.0	1.57	1.57	13.3	0.83	1.22
4	0.50	9.0	1.44	1.40	10.0	0.55	1.26
4	0.40	5.0	0.68	1.25	5.0	0.32	1.50
4	0.33	2.5	0.58	1.38	1.6	0.18	1.70
4	0.25	0.0	0.39	1.60	0.0	0.098	2.00
3	1.00	15.0	3.60	1.94	16.0	1.70	1.33
3	0.66	6.0	1.63	1.72	7.6	1.00	1.17
3	0.50	3.5	1.09	1.51	4.0	0.63	1.33
3	0.40	2.5	0.90	1.31	1.6	0.48	1.39
3	0.33	0.0	0.62	1.20	0.0	0.27	1.55
2.5	1.00	7.5	3.30	1.98	12.0	1.76	1.60
2.5	0.66	5.0	1.64	1.68	5.0	0.83	1.27
2.5	0.50	1.0	1.00	1.58	1.7	0.62	1.21
2.5	0.40	0.5	0.90	1.56	0.8	0.52	1.33
2.5	0.33	0 0	0.57	1.49	0.0	0.32	1.39
2	1.00	6.0	3.20	2.42	7.6	1.90	1.73
2	0.66	5.0	1.60	2.17	1.8	0.90	1.42
2	0.50	0.0	1.10	2.06	0.0	0.57	1.21
Single-Expansion Engine		—	—	1.75	—	—	1.35

Value of the Formulæ.

45) As the preceding formulæ disregard the finite length of connecting rod, the clearances, the compression, and the occurrence of the release before the end of stroke, the results derived from them will not it is true be quite accurate in practice, but they are nevertheless useful as guides in special cases.

Influence of the angle between the cranks.

46) Mess[rs] VAN VEEN and VAN ANDEL*) of the Dutch Navy have published a very simple and practical method of determining the receiver-pressures. They say that for the most usual working pressure of 5 atmos., if the receiver losses and the ratio of $\frac{max.}{mean}$ twisting moment are to be a minimum, we must make

$$\frac{V}{v} = \frac{8}{5};\ \frac{R}{v} = 1;\ \varepsilon = 0{,}304;\ \varepsilon_n = {}^5/_8 \text{ and } \varepsilon_g = 0{,}19.$$

Here however the angle between the cranks is assumed to be

*) Engineering 1882. I. p. 579.

not a right angle but greater than this in a certain proportion depending upon ε_n. The twisting moment is in general most uniform if the cranks are so placed that the L.P. cut-off occurs when the H.P. engine is on the centre. KÁŠ*) demonstrates very clearly how the angle between the cranks should be chosen in order to avoid the drop in the receiver with a given cylinder ratio and the most favourable L.P. cut-off ratio.

47) *Regarding all the principal points in the design of a compound collectively*, we arrive at the following. The *IHP*, the initial pressure p, and the total cut-off ratio ε_g being given, the diameter of the L.P. cylinder can be calculated. In determining the cylinder ratio $\frac{V}{v}$ it is to be borne in mind that the smaller this is taken, the smaller the weight of the engine comes out, as well as the first cost and the space in the ship, but that on the other hand, the stress on the rods is increased. In fixing upon the L.P. cut-off ratio ε_n, attention must be paid 1) to the loss due to sudden expansion which is a minimum when $\varepsilon_n = \frac{v}{V}$ so far as conditions of volume are concerned, but in any case is the smaller the closer ε_n approaches this value, 2) to the smallest stress on the rods, and 3) to the most uniform twisting moment. It is not in general possible to find a value of ε_n which fulfils all the three conditions, so that a selection must be made from among its values according to which condition is the most important. **Collation of the conditions for the design of a two-cylinder compound.**

48) In practice the cylinder-ratio $\frac{V}{v}$ is often made to depend on the working pressure p_x and we find for mercantile engines with **Empirical choice of a cylinder-ratio.**

p_x below 5.0	atmos.	$\frac{V}{v} = 3.00$
„ $= 5.0$	„	„ $= 3.50$
„ $= 5.5$	„	„ $= 3.75$
„ $= 6.0$	„	„ $= 4.00$
„ $= 6.5$	„	„ $= 4.50$

In naval engines the ratio is invariably smaller for the reason given in § 52, 5. The total ratio of cut-off ε_g also often bears a certain relation to the working pressure and in general

$$\varepsilon_g = \frac{4}{5\,p_x}$$

*) Berg- und Hüttenmännisches Jahrbuch der k. k. Bergakademie. Vienna. 1880. Vol. 28.

If these proportions are observed an expansion-slide for the L.P. is usually not required. But if the drop is to be avoided as far as possible the cylinder-ratio must be smaller and an L.P. expansion-slide fitted.

Concluding Remarks.

49) *Comparing finally the results arrived at by help of the foregoing investigations with the results of a single-expansion engine* we may say briefly that as against the expansion work of a single-expansion engine, the compound exhibits a certain mostly unavoidable loss of effect due to drop and to cooling and resistance in the passages as well as in the receiver, it occupies a greater space, it is heavier by the weight of the H.P. cylinder, and its first cost is higher. On the other hand, the stress on the rods is less in the compound, so that the bed-plate, columns, and rods can be lighter. When $\frac{V}{v}$ and ε_n are properly chosen, the twisting moment is more uniform than in the single-expansion engine and therefore shaft fractures (see § 42, 7) are less to be feared; and finally the consumption is much lower than in the single-expansion engine because the steam is better utilized (see § 45, 20 and 21).

§ 51.

Calculation of Three-cylinder Compounds.

Diameter of the L.P. cylinder.

1) **I. Determination of the cylinder diameters.** Three-cylinder compounds are hardly ever built now. The calculation is precisely the same as for a two-cylinder compound. If in the preliminary calculation for a compound engine the diameter of the L.P. cylinder comes out so large that there are practical difficulties in the way of making it, or that it cannot conveniently be got into the ship, we are forced to divide its volume in two, and the three cylinder compound is the result. This engine usually replaces the two-cylinder one when the latter's calculated L.P. diameter is greater than 2.5 m. Many designers limit it to 2.3 m while others have gone as far as 3 m with most unfavourable results (compare § 45, 22).

Actual reduced mean pressure.

2) In calculating the diameter of the L.P. cylinder, the reduced mean pressure can be assumed exactly as for a two-cylinder compound. The table below contains the trial-trip reduced mean pressures of two naval and two mercantile three-cylinder compounds which will serve as guides.

	Naval Engines		Mercantile Engines	
1	2	3	4	5
Name	"Olga"	"Leipzig"	"Elbe"	"Hammonia"
Working pressure in atmos.	5	5	5.27	5.62
Cylinder-ratio $\frac{V}{v}$	2.91	2.86	4.01	3.33
Actual reduced mean pressure in atmos.	1.98	1.86	1.93	1.85

3) For three-cylinder compounds again, besides deciding upon the positions of the cranks, it is essential to make a correct choice of a cylinder-ratio and an L.P. cut-off ratio. Having got out the ideal indicator diagrams, those of the loads including inertia, and of the twisting moments can be easily drawn as explained in § 48 for single-expansion engines. As it would take us too far to work out the cylinder-ratio and L.P. cut-off for all the four conditions of avoidance of losses of effect, equal work in all three-cylinders, minimum rod stresses, and uniform twisting moment — for each of the customary crank-arrangements detailed in § 45, 26 to 29, the following investigations will be confined to the theoretical indicator diagram of a three-cylinder compound with cranks at 120^0 and the H.P. crank leading. From this the most usual arrangement of cranks, the corresponding values for other arrangements and sequences of cranks can be derived without difficulty and it is obvious from the remarks upon the design of a two-cylinder compound in § 50, what calculations will lead to the same end for a three-cylinder one. These calculations are it is true, much more complicated, because the theoretical as well as the actual indicator cards of the two L.P. cylinders are scarcely ever equal. For this reason the combined indicator diagram is not given but, for greater clearness, only the theoretical indicator diagram for each cylinder separately. Diagrams.

4) **II. Construction of the theoretical indicator diagrams.** Assuming infinite connecting rods and neglecting clearances, there are three points of view from which to regard the distribution of the steam in a three-cylinder compound with cranks at 120^0, according as Construction of the theoretical indicator diagrams.

a) both L. P. cylinders are still in communication with the receiver (see § 45, 30) at the moment of H.P. release, i. e. their cut-off ratio $\varepsilon_{n} > 0.75$;

b) only one L.P. cylinder still communicates with the receiver at H.P. release, i. e. ε_n lies between *0.25* and *0.75*;

c) both L.P's. have cut off at H.P. release, i. e. $\varepsilon_n < 0.25$

Cut-off ratio > 0.75.

5) a. **L.P. cut-off ratio greater than 0.75.** A mere inspection of the positions of the cranks (Fig. 8, Pl. 15) at the H.P. release shews that the distribution of the steam in this case is very unfavourable. Calling the H.P. crank OA, the second L.P. crank OB and the first OC, the steam will pass from the H.P. cylinder through the receiver into the first L.P. when its piston is at quarter stroke, but not into the second until its piston is at three quarter stroke, i. e. just before cut-off. Consequently the work in the two L.P. cylinders is very unequal and one of them is practically of very little use. Further investigation of this case is therefore unnecessary as it never occurs in a well designed engine and all three-cylinder compounds have $\varepsilon_n < 0.75$.

Cut-off ratio between 0.25 and 0.75.

6) b. **The cut-off in both L.P. cylinders between 0.25 and 0.75** is the most usual case in practice. In the following all the former designations are retained except V which shall now signify the volume of *one* L.P. cylinder, so that the total cut-off ratio ε_g now becomes

$$\varepsilon_g = \frac{\varepsilon v}{2 V}$$

Here again as in § 50, 5 and 17, the relations between the piston and crank positions are to be evolved upon the assumption of an infinite connecting rod. In order to shorten the reasoning which is quite analogous to that for two-cylinder compounds, only the principal features of the indicator diagrams will be determined. The formulæ for the ordinates of the various curves can be easily derived from them by the foregoing.

Values of m' and m''.

7) In Fig. 9, Pl. 15 let OA be the position of the H.P. crank at the moment of cut-off in L.P. cylinder I at the crank position OC corresponding to the $\angle MOC = \beta$ and let OB be the simultaneous position of L.P. crank II. Then MF is the travel of L.P. piston I, ND that of the H.P. piston, and consequently MD that portion of the length of the cylinder which is in communication with the receiver; finally ME is the distance of L.P. piston II from the end of its stroke. Therefore we have

$$\frac{MF}{MN} = \varepsilon_n = \frac{1 - \cos\beta}{2}$$

$$\cos\beta = 1 - 2\,\varepsilon_n$$

$$\sin\beta = \sqrt{1 - \cos^2\beta} = 2\,\varepsilon_n \sqrt{\frac{1}{\varepsilon_n} - 1}$$

further

$$\frac{MD}{MN} = \frac{1 + \cos(\beta - 60^0)}{2}$$

$$= \frac{1 + {}^1/_2 \cos\beta + {}^1/_2 \sqrt{3} \times \sin\beta}{2}$$

$$= \frac{1 + {}^1/_2 (1 - 2\,\varepsilon_n) + {}^1/_2 \sqrt{3}\left(2\,\varepsilon_n \sqrt{\frac{1}{\varepsilon_n} - 1}\right)}{2}$$

$$= \frac{3 - 2\,\varepsilon_n + 2\,\varepsilon_n \sqrt{3\left(\frac{1}{\varepsilon_n} - 1\right)}}{4} = m'$$

$$1 - m' = \frac{1 + 2\,\varepsilon_n - 2\,\varepsilon_n \sqrt{3\left(\frac{1}{\varepsilon_n} - 1\right)}}{4}$$

and finally

$$\frac{ME}{MN} = \frac{1 + \cos(\beta + 60^0)}{2}$$

$$= \frac{1 + {}^1/_2 \cos\beta - {}^1/_2 \sqrt{3} \times \sin\beta}{2}$$

$$= \frac{1 + {}^1/_2 (1 - 2\,\varepsilon_n) - {}^1/_2 \sqrt{3}\left(2\,\varepsilon_n \sqrt{\frac{1}{\varepsilon_n} - 1}\right)}{2}$$

$$= \frac{3 - 2\,\varepsilon_n - 2\,\varepsilon_n \sqrt{3\left(\frac{1}{\varepsilon_n} - 1\right)}}{4} = m''$$

$$1 - m'' = \frac{1 + 2\,\varepsilon_n + 2\,\varepsilon_n \sqrt{3\left(\frac{1}{\varepsilon_n} - 1\right)}}{4}$$

The following table contains the values of m' and $1 - m'$ also m'' and $1 - m''$ for various L.P. cut-off ratios ε_n.

ε_n	0.25	0.30	0.35	0.40	0.45	0.50	0.55	0.60	0.65	0.70	0.75
$\frac{1}{\varepsilon_n}$	4.00	3.33	2.86	2.50	2.22	2.00	1.82	1.67	1.54	1.43	1.33
m'	1.000	0.997	0.988	0.974	0.955	0.933	0.906	0.875	0.838	0.797	0.748
$1 - m'$.	0.000	0.003	0.012	0.026	0.045	0.067	6.094	0.125	0.162	0.203	0.252
m''	0.250	0.203	0.162	0.126	0.094	0.067	0.044	0.026	0.010	0.003	0.000
$1 - m''$.	0.750	0.797	0.838	0.874	0.906	0.933	0.956	0.974	0.990	0.997	1.000

Pressure at cut-off in L.P. cyl. I.

8) In the following calculation of the pressure at the principal points of the stroke (Figs. 10 to 12, Pl. 15) the cut-off ratio and terminal pressure are assumed to be the same in both L.P. cylinders. Then in each of these cylinders the pressure at cut-off is $\frac{p\,\varepsilon_g}{\varepsilon_n}$ and the terminal pressure $p\,\varepsilon_g$. Regarding first the state of affairs in L.P. cylinder I, it appears that at the moment of its cut-off the volume of steam enclosed in the receiver is

$$R + m'\,v + \varepsilon_n V$$

and the pressure in the cylinder (L.P. I) is by the hypothesis

$$\frac{p\,\varepsilon_g}{\varepsilon_n} = Q_, H_, = Q_{,,} H_{,,} = Q\,H$$

Receiver pressure immediately ofter the H.P. exhaust p_r

9) When the L.P. piston I is at $^1/_4$ stroke, the H.P. exhaust takes place, and the volume in the receiver is

$$R + v + \frac{V}{4},$$

the pressure p_r being therefore

$$\frac{p\,\varepsilon_g}{\varepsilon_n}\left(R + m'\,v + \varepsilon_n\,V\right) = p_r\left(R + v + \frac{V}{4}\right)$$

$$p_r = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{R + m'\,v + \varepsilon_n\,V}{R + v + \frac{V}{4}}\right]$$

$$p_r = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + m' + \varepsilon_n \frac{V}{v}}{\frac{R}{v} + 1 + \frac{1}{4}\,\frac{V}{v}}\right] = L\,K = L_, K_, \quad \ldots . \; (255)$$

Receiver pressure immediately before the H.P. exhaust p''_r

10) Immediately before the H.P. exhaust, the receiver contains a volume of steam $R + \frac{V}{4}$, the pressure of which shall be called p''_r and with this volume the H.P. exhaust steam of volume v and pressure $p\,\varepsilon$ is combined, consequently the receiver pressure must rise from p''_r to p_r while simultaneously the receiver volume is increased to $R + \frac{V}{4} + v$. We have therefore

$$p''_r\left(V + \frac{V}{4}\right) + p\,\varepsilon\,v = p_r\left(R + v + \frac{V}{4}\right)$$

$$p''_r\left(R + \frac{V}{4}\right) + p\,\varepsilon\,v = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{R + m'\,v + \varepsilon_n V}{R + v + \frac{V}{4}}\right]\left(R + v + \frac{V}{4}\right)$$

$$p''_r = \frac{\frac{p\,\varepsilon_g}{\varepsilon_n}\left(R + m'\,v + \varepsilon_n\,V\right) - p\,\varepsilon\,v}{R + \frac{V}{4}}$$

but as $\varepsilon = \frac{2\,\varepsilon_g\,V}{v}$ and therefore $p\,\varepsilon = \frac{2\,p\,\varepsilon_g\,V}{v}$, it follows that

$$p_r'' = p\,\varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left(R + m'v\right) + V - 2V}{R + \frac{V}{4}} \right]$$

$$p_r'' = p\,\varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left(\frac{R}{v} + m'\right) - \frac{V}{v}}{\frac{R}{v} + \frac{1}{4}\frac{V}{v}} \right] = A\,F = L_,\,F_, \quad \ldots \; (256)$$

11) After the cut-off in L.P. cylinder I, the steam remaining in the receiver is compressed by the H.P. piston until the beginning of the next stroke in L.P. cylinder II. The volume of steam remaining in the receiver at the cut-off in L.P. cylinder I is **Initial pressure in L.P. cylinder II. p'_{II}**

$$R + m'\,v.$$

At the beginning of the stroke of L.P. II the above volume is compressed by the H.P. piston to $R + 0.75\,v$ and the pressure p'_{II} of this steam will be

$$\frac{p\,\varepsilon_g}{\varepsilon_n}(R + m'\,v) = p'_{II}(R + 0{,}75\,v)$$

$$p'_{II} = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{R + m'\,v}{R + 0{,}75\,v}\right]$$

$$p'_{II} = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + m'}{\frac{R}{v} + \frac{3}{4}}\right] = V\,J = V_,\,J_, = V_{,,}\,J_{,,} \quad \ldots \; (257)$$

12) This steam now acts during a quarter of the stroke both behind L.P. piston II and in front of the H.P. piston until L.P. I begins its next stroke. The volume then present in the receiver is **Initial pressure in L.P. cylinder I. p'_I**

$$R + 0{,}25\,(v + V)$$

and the pressure p'_I is

$$\frac{p\,\varepsilon_g}{\varepsilon_n}(R + m'\,v) = p'_I\left[R + 0{,}25\,(v + V)\right]$$

$$p'_I = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{R + m'\,v}{R + 0{,}25\,(v + V)}\right]$$

$$p'_I = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + m'}{\frac{R}{v} + \frac{1}{4}\left(1 + \frac{V}{v}\right)}\right] = T\,S = T_,\,S_, = T_{,,}\,S_{,,} = B_,\,R_, \quad (258)$$

13) The steam of the pressure above calculated drives both the L.P. pistons and exerts a back-pressure in the H.P. cylinder — until the cut-off in L.P. II; immediately before this cut-off therefore, the receiver volume is **Receiver pressure at the several cut-offs in the L.P. cylinders p_1**

$$R + \varepsilon_n V + (1 - m') V + m'' v$$

and the pressure of this steam p_1, which is obtained from

$$\frac{p\,\varepsilon_g}{\varepsilon_n}(R + m' v) = p_1 (R + \varepsilon_n V + (1 - m') V + m'' v)$$

$$\text{is } p_1 = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{R + m' v}{R + \varepsilon_n V + (1 - m') V + m'' v}\right]$$

$$p_1 = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + m'}{\frac{R}{v} + \frac{V}{v}(\varepsilon_n + 1 - m') + m''}\right] = Q_{\prime\prime}\,H_{\prime\prime}\;.\;.\;.\;(259)$$

In the case, here assumed, of equal cut-offs in both L.P. cylinders the expression in the bracket = 1 (see the last equation in 8). With this all the essential points of the diagram are determined and it can be drawn as soon as the expressions for the ordinates of the several curves have been worked out analogously to those in § 50.

Cut-off ratio < 0.25

14) c. **The cut-off ratio in both L.P. cylinders less than 0.25** is a case that only occurs very rarely in practice and then only when the engine is working continuously at reduced power, with the L.P. expansion gear (if fitted) in operation. In engines where the combined volume of the two L.P. cylinders is at least four times as great as the volume of the H.P. cylinder, i. e. where $2\,V > 4\,v$ the cut-off in both L.P's. must in certain cases be earlier than 0.25 to prevent any considerable drop.

Values of m' and m''.

15) If OA (Fig. 13, Pl. 15) represents the position of the H.P. crank when the L.P. crank OC is at the angle $MOC = \beta$ and the cut-off occurs in L.P. I, also OB the position of L.P. crank II, then MF is the travel of L.P. piston I from the beginning of its stroke, ND the small distance of the H.P. piston from the end of its stroke and at the same time that portion of the length of the cylinder which is in communication with the receiver, finally ME is the distance of L.P. piston II from the end of its stroke. Therefore we have

$$\frac{MF}{MN} = \varepsilon_n = \frac{1 - \cos\beta}{2}$$

$\cos\beta = 1 - 2\,\varepsilon_n$ and by the preceding

$$\sin\beta = 2\,\varepsilon_n\sqrt{\frac{1}{\varepsilon_n} - 1};$$

further

$$\frac{ND}{MN} = \frac{1 + \cos(120^0 + \beta)}{2}$$

$$= \frac{1 - {}^1/_2\cos\beta - {}^1/_2\sqrt{3}\sin\beta}{2}$$

$$= \frac{1 - \frac{1 - 2\,\varepsilon_n}{2} - 2\,\varepsilon_n \sqrt{3}\sqrt{\frac{1}{\varepsilon_n} - 1}}{2}$$

$$= \frac{1 + 2\,\varepsilon_n - 2\,\varepsilon_n \sqrt{3\left(\frac{1}{\varepsilon_n} - 1\right)}}{4} = m';$$

and finally

$$\frac{ME}{MN} = \frac{1 - \cos(120^0 - \beta)}{2}$$

$$= \frac{1 + {}^1/_2 \cos\beta - {}^1/_2 \sqrt{3}\sin\beta}{2}$$

$$= \frac{1 + \frac{1 - 2\,\varepsilon_n}{2} - 2\,\varepsilon_n \sqrt{3}\sqrt{\frac{1}{\varepsilon_n} - 1}}{2}$$

$$= \frac{3 - 2\,\varepsilon_n - 2\,\varepsilon \sqrt{3\left(\frac{1}{\varepsilon_n} - 1\right)}}{4} = m''$$

The following table contains the values of m' and m'' for various ratios of cut-off in the L.P. cylinders

ε_n	0.100	0.125	0.150	0.175	0.200
$\frac{1}{\varepsilon_n}$	10.00	8.00	6.67	5.71	5.00
m'	0.040	0.026	0.016	0.008	0.003
m''	0.440	0.400	0.365	0.330	0.303

16) Again assuming the cut-off ratios and terminal pressures to be the same in both L.P. cylinders (Figs. 14 to 16, Pl. 15) then the pressure at cut-off in both is $\frac{p\,\varepsilon_g}{\varepsilon_n}$ and the terminal pressure $p\,\varepsilon_g$. Immediately before the cut-off in L.P. I, the volume of steam enclosed in the receiver is **Initial pressure in L.P. cylinder I. p'_I.**

$$R + m'\,v + \varepsilon_n\,V$$

and as the volume of this at the beginning of the stroke in L.P. I was

$$R + \frac{v}{4}$$

it follows that its pressure p'_I must have been

$$\frac{p\,\varepsilon_g}{\varepsilon_n}\left(R + m'\,v + \varepsilon_n\,V\right) = p'_I\left(R + \frac{v}{4}\right)$$

$$p'_I = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{R + \varepsilon_n\,V + m'\,v}{R + \frac{v}{4}}\right]$$

$$p'_I = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + \varepsilon_n \frac{V}{v} + m'}{\frac{R}{v} + \frac{1}{4}}\right] = T\,S = T_, S_, = B_, R_, \quad . \; . \quad (260)$$

Receiver pressure immediately before the H.P. exhaust p''_r.

17) Immediately after the cut-off in L.P. I, the volume $R + m' v$ remains in the receiver and at the end of the stroke of the H.P. piston is compressed by it to R. The back-pressure p''_r against the H.P. piston at the end of the stroke is therefore

$$\frac{p\,\varepsilon_g}{\varepsilon_n}\left(R + m' v\right) = p''_r R$$

$$p''_r = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{R + m' v}{R}\right]$$

$$p''_r = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + m'}{\frac{R}{v}}\right] = A\,F = L_, F_, \quad . \; . \; . \; . \; . \; . \quad (261)$$

Receiver pressure immediately after the H.P exhaust p_r.

18) The H.P. exhaust steam of volume v and pressure $p\,\varepsilon$ passes into the receiver and the pressure in this rises from $p''_r = A\,F$ to $p_r = L\,K = L_, K_,$ which pressure is calculated as follows

$$p''_r R + p\,\varepsilon\,v = p_r (R + v)$$

$$p_r = \frac{p''_r + p\,\varepsilon\,v}{R + v}$$

and as $\varepsilon = \frac{2\,\varepsilon_g\,V}{v}$, we get, after substituting the value of p''_r

$$p_r = \frac{\frac{p\,\varepsilon_g}{\varepsilon_n}\left(R + m' v\right) + 2\,p\,\varepsilon_g \frac{V}{v} v}{R + v}$$

$$p_r = p\,\varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left(R + m' v\right) + 2\,V}{R + v}\right]$$

$$p_r = p\,\varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left(\frac{R}{v} + m'\right) + 2\,\frac{V}{v}}{\frac{R}{v} + 1}\right] = L\,K = L_, K_, \quad . \; . \; . \quad (262)$$

Initial pressure in L.P. cyl. II p'_{II}.

19) The steam of this pressure is compressed by the H.P. piston during one fourth of its stroke until the L.P. piston II passes the centre and the steam can get into that cylinder. The volume of steam enclosed in the receiver at the beginning of the stroke in L.P. II is therefore

$$R + 0{,}75\,v$$

and its pressure p'_{II}, the initial pressure of L.P. II is

$$p\,\varepsilon_g\left[\frac{1}{\varepsilon_n}\left(R + m' v\right) + 2\,V\right] = p'_{II}(R + 0{,}75\,v)$$

$$p'_{II} = p\,\varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left\{R + m'v\right\} + 2V}{R + 0{,}75\,v} \right]$$

$$p'_{II} = p\,\varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left\{\frac{R}{v} + m'\right\} + 2\frac{V}{v}}{\frac{R}{v} + \frac{4}{3}} \right] = V\mathfrak{J} = V_{,}\mathfrak{J}_{,} = V_{,,}\mathfrak{J}_{,,}. \quad (263)$$

20) The steam at this pressure fills L.P. cylinder II up to the ε_n th part of its stroke and its volume immediately before cut-off is Receiver pressure at cut-off in both L.P. cylinders p_1.

$$R + m''v + \varepsilon_n V,$$

so that its pressure is

$$p\,\varepsilon_g \left[\frac{1}{\varepsilon_n}\left(R + m'v\right) + 2V\right] = p_1\,(R + m''v + \varepsilon_n V)$$

$$p_1 = p\,\varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left(R + m'v\right) + 2V}{R + m''v + \varepsilon_n V} \right]$$

$$p_1 = \frac{p\,\varepsilon_g}{\varepsilon_n} \left[\frac{\frac{R}{v} + m' + 2\,\varepsilon_n \frac{V}{v}}{\frac{R}{v} + m'' + \varepsilon_n \frac{V}{v}} \right] = Q_{,,}\,H_{,,} \quad \ldots\ldots (264)$$

In this equation again the expression in the bracket = 1 for equal cut-offs in both L.P. cylinders, analogously to Eq. 259. The principal points of the diagram are thus determined and in order to draw it exactly it only remains to compute the ordinates of the several curves as before.

21) All the formulæ relating to the diagrams are recapitulated below, viz. Collation of the formulæ.

α) H.P. initial pressure

$$AC = p \text{ in atmos. absolute,}$$

β) Terminal pressure in both L.P. cylinders

$$N_{,}G_{,} = N_{,,}G_{,,} = p\,\varepsilon_g$$

γ) Back-pressure in condenser

$$B_{,}N_{,} = B_{,,}N_{,,} = a$$

δ) Terminal pressure in H.P.

$$LE = p\,\varepsilon = \frac{2\,p\,\varepsilon_g\,V}{v}$$

ε) Receiver pressure immediately after H.P. exhaust

$$LK = L_{,}K_{,} = p_r = p\,\varepsilon_g \left[\frac{\frac{1}{\varepsilon_n}\left(\frac{R}{v} + m'\right) + 2\frac{V}{v}}{\frac{R}{v} + 1} \right] \text{ for } \varepsilon < 0{,}25$$

$$L\,K = L_{,}\,K_{,} = p_r = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + m' + \varepsilon_n \frac{V}{v}}{\frac{R}{v} + 1 + \frac{1}{4}\frac{V}{v}}\right] \quad \text{for } \varepsilon_n = 0.25 \text{ to } 0.75$$

ζ) Receiver pressure immediately before cut-off in both L.P. cylinders

$$Q\,H = Q_{,}\,H_{,} = Q_{,,}\,H_{,,} = p = \frac{p\,\varepsilon_g}{\varepsilon_n}$$

η) Initial pressure in L.P. cylinder I

$$T\,S = T_{,}\,S_{,} = B_{,}\,R_{,} = p'_I = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + \varepsilon_n \frac{V}{v} + m'}{\frac{R}{v} + \frac{1}{4}}\right] \quad \text{for } \varepsilon_n < 0.25$$

$$T\,S = T_{,}\,S_{,} = T_{,,}\,S_{,,} = B_{,}\,R_{,} = p'_I = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + m'}{\frac{R}{v} + \frac{1}{4}\left(1 + \frac{V}{v}\right)}\right] \quad \text{for } \varepsilon_n = 0.25 \text{ to } 0.75$$

ϑ) Initial pressure in L.P. cylinder II

$$V\,J = V_{,}\,J_{,} = V_{,,}\,J_{,,} = p''_{II} = p\,\varepsilon_g\left[\frac{\frac{1}{\varepsilon_n}\left(\frac{R}{v} + m'\right) + 2\frac{V}{v}}{\frac{R}{v} + \frac{3}{4}}\right] \quad \text{for } \varepsilon_n < 0.25$$

$$V\,J = V_{,}\,J_{,} = V_{,,}\,J_{,,} = p''_{II} = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + m'}{\frac{R}{v} + \frac{3}{4}}\right] \quad \text{for } \varepsilon_n = 0.25 \text{ to } 0.75$$

ι) Receiver pressure immediately before the H.P. exhaust

$$A\,F = L_{,}\,F_{,} = p''_r = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + m'}{\frac{R}{v}}\right] \quad \text{for } \varepsilon_n < 0.25$$

$$A\,F = L_{,}\,F_{,} = p''_r = p\,\varepsilon_g\left[\frac{\frac{1}{\varepsilon_n}\left(\frac{R}{v} + m'\right) - \frac{V}{v}}{\frac{R}{v} + \frac{1}{4}\frac{V}{v}}\right] \quad \text{for } \varepsilon_n = 0.25 \text{ to } 0.75$$

Cylinder-ratio $\frac{V}{v}$

22) **III. Determination of the cylinder-ratio $\frac{V}{v}$.** All the above formulæ are evolved under the assumption in 8) viz. that the pressure at the beginning of the expansion is the same in both L.P. cylinders.

This pressure for L.P. cylinder II is determined by Eqq. 259 and 264 for the two cases when the cut-off ratio $\varepsilon_n < 0.25$ and lies between 0.25 and 0.75 respectively. Equating this pressure to $\frac{p\,\varepsilon_g}{\varepsilon_n}$ the corresponding pressure in L.P. cylinder I, we can as shewn below, calculate from these equations the cylinder volumes which for equal L.P. cut-offs ε_n will give equal terminal pressures.

$\varepsilon_n < 0.25$

$$\frac{p\,\varepsilon_g}{\varepsilon_n} = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + m' + 2\,\varepsilon_n \frac{V}{v}}{\frac{R}{v} + m'' + \varepsilon_n \frac{V}{v}}\right]$$

$$1 = \frac{\frac{R}{v} + m' + 2\,\varepsilon_n \frac{V}{v}}{\frac{R}{v} + m'' + \varepsilon_n \frac{V}{v}}$$

$$\frac{R}{v} + m'' + \varepsilon_n \frac{V}{v} = \frac{R}{v} + m' + 2\,\varepsilon_n \frac{V}{v}$$

$$m'' - m' = \varepsilon_n \frac{V}{v}$$

$$\frac{V}{v} = \frac{m'' - m'}{\varepsilon_n} \quad \ldots (265)$$

$\varepsilon_n = 0.25$ to 0.75

$$\frac{p\,\varepsilon_g}{\varepsilon_n} = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v} + m'}{\frac{R}{v} + \frac{V}{v}\left(\varepsilon_n + 1 - m'\right) + m''}\right]$$

$$1 = \frac{\frac{R}{v} + m'}{\frac{R}{v} + \frac{V}{v}\left(\varepsilon_n + 1 - m'\right) + m''}$$

$$\frac{R}{v} + \frac{V}{v}\left(\varepsilon_n + 1 - m'\right) + m'' = \frac{R}{v} + m'$$

$$\frac{V}{v}\left(\varepsilon_n + 1 - m'\right) = m' - m''$$

$$\frac{V}{v} = \frac{m' - m''}{\varepsilon_n + 1 - m'} \quad \ldots (266)$$

The following table is calculated by these equations.

Table of Cylinder-ratios for Three-cylinder Compounds having their L.P. cut-offs and terminal pressures equal.

ε_n	0.100	0.125	0.150	0.175	0.200	0.250	0.300	0.350	0.400	0.450	0.500	0.550	0.600	0.650	0.700	0.750
$\frac{1}{\varepsilon_n}$	10.00	8.00	6.67	5.71	5.00	4.00	3.33	2.86	2.50	2.22	2.00	1.82	1.67	1.54	1.43	1.33
$\frac{V}{v}$	4.00	2.99	2.327	1.85	1.50	3.00	2.62	2.28	1.99	1.75	1.52	1.34	1.17	1.02	0.88	0.75

23) With other cylinder-ratios than those in the table the L.P. cut-off ratios and terminal pressures are not equal and it therefore happens that in most three-cylinder compounds with equal L.P. cut-off ratios the L.P. terminal pressures are unequal and vice versâ. We will now investigate for various cylinder and receiver ratios Cut-off ratios and terminal pressures.

a) what relation must exist between the L.P. cut-off ratios, to give equal terminal pressures;

b) what proportion the terminal pressures bear to each other when the cut-offs in both L.P's. are equal.

Equal L.P. terminal pressures.

24) a. **The L.P. terminal pressures are to be equal,** then the cut-off ratios are different.

Let ε_n = the cut-off ratio in L.P. I,
and ε_n' = " " " " " " II.

The development of the formulæ for the various points of the diagrams remains exactly the same as before only that the value of p_1 is changed, as instead of the former m' and m'', we have $m_{\prime}'$ and and $m_{\prime}''$ corresponding to the cut-off ratio ε_n'. The pressure at cut-off in L.P. II is then

$$p_1 = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v}+m'+2\,\varepsilon_n\frac{V}{v}}{\frac{R}{v}+m_{\prime}''+\varepsilon_n'\frac{V}{v}}\right] \text{ for } \varepsilon_n < 0.25$$

$$p_1 = \frac{p\,\varepsilon_g}{\varepsilon_n}\left[\frac{\frac{R}{v}+m'}{\frac{R}{v}+\frac{V}{v}\left(\varepsilon_n'+1-m_{\prime}'\right)+m_{\prime}''}\right] \text{ for } \varepsilon_n = 0.25 \text{ to } 0.75$$

The terminal pressure in this cylinder must therefore be

$$p_1\,\varepsilon_n' = p\,\varepsilon_g\,\varepsilon_n'\left[\frac{\frac{1}{\varepsilon_n}\left(\frac{R}{v}+m'\right)+2\frac{V}{v}}{\frac{R}{v}+m_{\prime}''+\varepsilon_n'\frac{V}{v}}\right] \text{ for } \varepsilon_n < 0.25$$

$$p_1\,\varepsilon_n' = p\,\varepsilon_g\,\frac{\varepsilon_n'}{\varepsilon_n}\left[\frac{\frac{R}{v}+m_{\prime}'}{\frac{R}{v}+\frac{V}{v}\left(\varepsilon_n'+1-m_{\prime}'\right)+m_{\prime}''}\right] \text{ for } \varepsilon_n = 0.25 \text{ to } 0.75.$$

But this pressure is by the hypothesis $= p\,\varepsilon_g$, so that we get

$$p\,\varepsilon_g = p\,\varepsilon_g\,\varepsilon_n'\left[\frac{\frac{1}{\varepsilon_n}\left(\frac{R}{v}+m'\right)+2\frac{V}{v}}{\frac{R}{v}+m_{\prime}''+\varepsilon_n'\frac{V}{v}}\right] \text{ for } \varepsilon_n < 0.25$$

$$p\,\varepsilon_g = p\,\varepsilon_g\,\frac{\varepsilon_n'}{\varepsilon_n}\left[\frac{\frac{R}{v}+m'}{\frac{R}{v}+\frac{V}{v}\left(\varepsilon_n'+1-m_{\prime}'\right)+m_{\prime}''}\right] \text{ for } \varepsilon_n = 0.25 \text{ to } 0.75$$

$$1 = \varepsilon_n' \left[\frac{\frac{1}{\varepsilon_n}\left(\frac{R}{v} + m'\right) + 2\,\frac{V}{v}}{\frac{R}{v} + m_{,}'' + \varepsilon_n' \frac{V}{v}} \right] \text{ for } \varepsilon_n < 0.25$$

$$\frac{\varepsilon_n'}{\varepsilon_n} = \frac{\frac{R}{v} + \frac{V}{v}\left(\varepsilon_n' + 1 - m_{,}'\right) + m_{,}''}{\frac{R}{v} + m'} \text{ for } \varepsilon_n = 0.25 \text{ to } 0.75$$

$$\frac{1}{\varepsilon_n}\left(\frac{R}{v} + m_{,}''\right) + \frac{V}{v} = \frac{1}{\varepsilon_n}\left(\frac{R}{v} + m'\right) + 2\,\frac{V}{v} \text{ for } \varepsilon_n < 0.25$$

$$\frac{1}{\varepsilon_n}\left(\frac{R}{v} + m'\right) = \frac{1}{\varepsilon_n}\left[\frac{R}{v} + \frac{V}{v}\left(1 - m_{,}'\right) + m_{,}''\right] + \frac{V}{v} \text{ for } \varepsilon_n = 0.25 \text{ to } 0.75$$

for $\varepsilon_n < 0.25$	for $\varepsilon_n = 0.25$ to 0.75
(267) $\dfrac{\varepsilon_n}{\varepsilon_n'} = \dfrac{\frac{R}{v} + m' + \varepsilon_n \frac{V}{v}}{\frac{R}{v} + m_{,}''}$	$\dfrac{\varepsilon_n}{\varepsilon_n'} = \dfrac{\frac{R}{v} + m' - \varepsilon_n \frac{V}{v}}{\frac{R}{v} + \frac{V}{v}\left(1 - m_{,}\right) + m_{,}''}$ (268)
$\varepsilon_n' = \dfrac{\frac{R}{v} + m_{,}''}{\frac{1}{\varepsilon_n}\left(\frac{R}{v} + m'\right) + \frac{V}{v}}$	$\varepsilon_n' = \dfrac{\frac{R}{v} + \frac{V}{v}\left(1 - m_{,}\right) + m_{,}''}{\frac{1}{\varepsilon_n}\left(\frac{R}{v} + m'\right) - \frac{V}{v}}$

If ε_n and ε_n' do not differ much we can put m' for $m_{,}'$ and m'' for $m_{,}''$ and thus obtain values of ε_n' sufficiently accurate for practice which can easily be evolved from the foregoing formulæ for any value of ε_n we please.

25) b. **If the cut-offs in the L.P. cylinders are to be equal** then the terminal pressures are different. **Equal cut-offs in the L.P.'s.**

Let p_I be the terminal pressure in L.P. I,
and p_{II} " " " " " " II.

We will also assume that the mean of these two pressures equals the theoretical terminal pressure $p\,\varepsilon_g$,

$$\frac{1}{2}\left(p_I + p_{II}\right) = p\,\varepsilon_g$$

Again in this assumed case of equal cut-offs in the L.P. cylinders the formulæ for the pressures in the cylinders and receivers remain the same as before except that the actual terminal

pressure p_I in L.P. I takes the place of the theoretical terminal pressure $p\,\varepsilon_g$. The pressure p_1 at cut-off in L.P. II is then

$$p_1 = p_I \left[\frac{\frac{1}{\varepsilon_n}\left(\frac{R}{v} + m'\right) + 2\frac{V}{v}}{\frac{R}{v} + m'' + \varepsilon_n \frac{V}{v}} \right] \text{ for } \varepsilon_n < 0.25$$

$$p_1 = p_I \frac{1}{\varepsilon_n} \left[\frac{\frac{R}{v} + m'}{\frac{R}{v} + \frac{V}{v}\left(\varepsilon_n + 1 = m'\right) + m''} \right] \text{ for } \varepsilon_n = 0.25 \text{ to } 0.75;$$

and as by the hypothesis this pressure $= p_{II}\frac{1}{\varepsilon_n}$, it follows that

$$p_{II}\frac{1}{\varepsilon_n} = p_I \left[\frac{\frac{1}{\varepsilon_n}\left(\frac{R}{v} + m'\right) + 2\frac{V}{v}}{\frac{R}{v} + m'' + \varepsilon_n \frac{V}{v}} \right] \text{ for } \varepsilon_n < 0.25$$

$$p_{II}\frac{1}{\varepsilon_n} = p_I \frac{1}{\varepsilon_n} \left[\frac{\frac{R}{v} + m'}{\frac{R}{v} + \frac{V}{v}\left(\varepsilon_n + 1 - m'\right) + m''} \right] \text{ for } \varepsilon_n = 0.25 \text{ to } 0.75$$

$$\frac{p_I}{p_{II}} = \frac{\frac{1}{\varepsilon_n}\left(\frac{R}{v} + m''\right) + \frac{V}{v}}{\frac{1}{\varepsilon_n}\left(\frac{R}{v} + m'\right) + 2\frac{V}{v}} \text{ for } \varepsilon_n < 0.25 \; . \; . \; . \; . \; . \; . \quad (269)$$

$$\frac{p_I}{p_{II}} = \frac{\frac{R}{v} + \frac{V}{v}\left(\varepsilon_n + 1 - m'\right) + m''}{\frac{R}{v} + m'} \text{ for } \varepsilon_n = 0.25 \text{ to } 0.75 \; . \; . \quad (270)$$

The relation of the terminal pressures p_I and p_{II} to each other being thus determined, the actual terminal pressure in each cylinder can be found by help of the equation $\frac{1}{2}(p_I + p_{II}) = p\,\varepsilon_g$

Table of the Cut-off ratios and Terminal Pressures in the L.P. Cylinders of Three-cylinder Compounds for various Cylinder and Receiver-ratios.

Cylinder-ratio $\frac{V}{v}$	1			2			3		
Receiver-ratio $\frac{R}{v}$	1	2	3	1	2	3	1	2	3
ε_n	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
ε'_n	0.126	0.114	0.109	0.116	0.109	0.106	0.107	0.104	0.103
$p_I : p_{II}$	1.279	1.152	1.092	1.138	1.081	1.060	1.061	1.040	1.030
ε_n	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2
ε'_n	0.216	0.209	0.206	0.185	0.191	0.194	0.162	0.177	0.183
$p_I : p_{II}$	1.071	1.041	1.028	0.944	0.964	0.973	0.864	0.906	0.929
ε_n	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
ε'_n	0.226	0.252	0.264	0.249	0.264	0.275	0.270	0.282	0.285
$p_I : p_{II}$	0.755	0.840	0.880	0.830	0.880	0.915	0.900	0.940	0.950
ε_n	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4
ε'_n	0.312	0.340	0.360	0.356	0.368	0.376	0.398	0.399	0.400
$p_I : p_{II}$	0.780	0.850	0.900	0.890	0.920	0.940	0.995	0.997	1.000
ε_n	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5
ε'_n	0.400	0.450	0.460	0.495	0.500	0.500	0.570	0.545	0.530
$p_I : p_{II}$	0.800	0.900	0.920	0.990	1.000	1.000	1.140	1.090	1.060
ε_n	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
ε'_n	0.558	0.576	0.582	0.678	0.648	0.636	0.780	0.720	0.690
$p_I : p_{II}$	0.930	0.960	0.970	1.130	1.080	1.060	1.300	1.200	1.150
ε_n	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7
ε'_n	0.742	0.728	0.721	0.910	0.840	0.798	1.092	0.952	0.882
$p_I : p_{II}$	1.060	1.040	1.030	1.300	1.200	1.140	1.560	1.360	1.260

26) The preceding table contains — for the most usual cylinder-ratios $\frac{V}{v}$ and various receiver-ratios $\frac{R}{v}$, **Table.**

1) for certain cut-off ratios ε_n in L.P. I, the necessary cut-off ratios ε'_n in L.P. II to produce equal terminal pressures p_I and p_{II};
2) the proportion between the terminal pressures p_I and p_{II} when the L.P. cut-offs are the same.

27) In conclusion we may remark that it is not unusual in practice to fix upon a cylinder-ratio dependent on the working **Practical choice of a cylinder-ratio.**

pressure p_x for three-cylinder compounds as well as two-cylinder ones; for instance in mercantile engines we find for

p_x not exceeding 6 atmos. . . . . $\frac{V}{v} = 3.0$

" " " 6.5 " . . . . " $= 3.4$

" " " 7 " . . . . " $= 3.7$

" " " 7.5 " . . . . " $= 4.0$

V being the combined volume of the two L.P. cylinders. Naval engines have mostly a smaller cylinder-ratio as instanced by the values given in 2) and for the reasons detailed in § 52, 5.

Schröter's method.

28) **IV. Graphic method of determining the steam volume** in the cylinders communicating with each other. The pressures in compound engines can be very much more easily calculated than as shewn in §§ 49 to 51 by a graphic method introduced by ZEUNER*) and further specially adapted to the present purpose by SCHRÖTER**). It will now be described as applied to a three-cylinder compound with cranks at 120° and otherwise of the following particulars, viz. $p = 7$ atmos. absolute, $a = 0.15$, $\frac{V}{v} = 4.2$, i. e. 2.1 for each of the L.P. cylinders, $\frac{R}{v} = 3.5$, $\varepsilon = 0.4$. The clearances are 5 % of the volume swept in the H.P. and 4 % in the L.P.'s. Further it is intended that there shall be no drop. The method is based upon

a) the construction of a piston diagram, whence is derived

b) " " " the theoretical indicator diagrams.

Construction of the piston diagram.

29) a. **The piston diagram** which forms the basis of the whole process, is got out as follows. On a horizontal line (Fig. 1, Pl. 16) the volumes to be filled by the steam are successively stepped off to any convenient scale, the volume swept by the H.P. piston being regarded as unity. In the present case therefore, beginning at the right hand, the order is — the volume of the H.P. cylinder, its clearance space, the volume of the receiver, the clearance space of L.P. I, the volume of L.P. I, the clearance space of L.P. II, and lastly its cylinder-volume. Among these quantities the receiver volume and the clearances are constant, the cylinder volumes are variable and the piston diagram represents the law of their variation. The next step is to draw a circle with the volume of each cylinder as diameter and divide the circumferences of these circles into any number of parts (here 24). A straight line AB of any convenient length (Fig. 1, Pl. 16) is now taken to represent the developed circumference of the circle and divided into the same number of parts, and the piston-

*) G. ZEUNER, Grundzüge der mechanischen Wärmetheorie. Edit. II. 1866. p. 200.

**) Zeitschrift des Vereins deutscher Ingenieure 1884. p. 191.

travels corresponding to these spots laid off horizontally to the left for the out-stroke, to the right for the in-stroke. In the present case the connecting rod is assumed to be infinite, but in investigating existing engines its length, i. e. the discrepancy between the piston-travels for top and bottom, can be taken account of, which is not worth while for a proposed new engine. The length of AB is thus quite indifferent so long as accuracy is preserved, only the horizontal measurements in the piston diagram being of importance for the construction of the indicator diagrams. Care must be taken to keep the proper relative positions of the pistons corresponding to the angles between the cranks, in this case 120^0 as shewn in the figure.

30) b. **The theoretical indicator-diagrams** are best drawn natural size, i. e. as they are produced by the indicator (see § 18, 15 and 18). Those given in the figure are shortened for want of space. It is advisable to take the L.P. pressure-scale $\frac{V}{v}$ times as coarse as that of the H.P., because then the areas of the diagrams can be immediately compared by means of the planimeter and the distribution of work in the cylinders directly judged of. The accompanying inconvenience of having to shorten or lengthen the piston-travels to suit the length of the indicator-card as shewn by the outside circles on the right in Fig. 1, Pl. 16, is worth putting up with. MARIOTTE'S line (§ 6, 16) serves as the expansion curve. The upper boundary line of the H.P. diagram 0 — 1 — 2 (Fig. 2, Pl. 16) is thus disposed of without further difficulty. **Construction of the indicator-diagrams.**

31) At the moment of communication between the H.P. cylinder and the receiver, the pressure in the latter must be the same as that at the point 2 (here 3 atmos.) and the two L.P's. must at this time have cut off. According to the piston-diagram we now have compression in the receiver until the point 3 where communication is established with L.P. cylinder I and the steam passes into it without drop. The ordinates of the curve 2—3 as well as those of the succeeding curves are determined by MARIOTTE'S law $pv = const.$, the abscissæ or volumes being taken directly from the piston-diagram. For instance the abscissa at B in the piston-diagram corresponding to the point 2 in the H.P. diagram is 135 mm long, the abscissa at C in the piston-diagram corresponding to the point 3 of the H.P. diagram measures 130 mm. As the pressure at point 2 is 3 atmos., if we call the pressure at point 3, x we get **Calculation of the pressures.**

$$135 \times 3 = 130\ x$$
$$x = 3.11 \text{ atmos.}$$

Compression curves.

32) The compression which lasts in the receiver till point 3, continues after the communication of the latter with L.P. cylinder I, as shewn by the dotted curve at C on the H.P. piston-diagram and only goes over into expansion at the point where the vertical is a tangent to this curve. Taking point 4 as a trial cut-off in L.P. cylinder I, the piece $3-4$ of the H.P. diagram corresponds to the piece $0—1'$ in L.P. I diagram (Fig. 3, Pl. 16). From point 4 to point 5 there is again a small amount of compression in the receiver until L.P. piston II reaches its dead point. The consequence of the accession of the volumes swept by L.P. piston II is at first a continuation of the compression, as the dotted curve at D in the H.P. diagram shews. During this compression which immediately changes to expansion, the portion 5—6 of the H.P. indicator diagram and the corresponding portion $0—1''$ of the L.P. II indicator diagram are described. At point 6 compression begins in the H.P. clearance. But at this time the receiver pressure is higher than the H.P. terminal pressure and therefore L.P. cylinder II must remain longer in communication with the receiver, until the right pressure (here 3 atmos.) is reached which occurs at point $2''$ in the L.P. II diagram. L.P. II now cuts off and the receiver remains isolated until the end of the stroke in the H.P. The ordinates of the curve 6—0 in the H.P. diagram are calculated according to the compression volumes exhibited in the H.P. piston-diagram. The compression curve 6—0 must reach the initial pressure at the point 0 if there is to be no drop. The expansion curves $1'-2'$ and $2''-2''_0$ in the L.P. diagrams are MARIOTTE'S lines. The compression curves of the L.P. diagrams $6'-0$ and $6''-0$ are most closely represented by adiabatics of index 1.2, bearing in mind what is said in § 18, 12. If the areas of the indicator diagrams thus produced differ considerably, then the admission line $0—1$ in the H.P. or that of L.P. I from 0 to $1'$ and with the latter also point 4 in the H.P. indicator diagram must be altered. The distances $0—1'$ and $0-2''$ or the admission lines of the two L.P. cylinders can be best decided on by help of the table on p. 481 according to the cylinder and receiver-ratios.

§ 52.

Calculation of Triple and Quadruple-expansion Engines.

Diameter of L.P. cylinder.

1) **I. Determination of the diameter of the L.P. cylinder.** The calculation of triple and quadruple-expansion engines is similar to that for compounds, i. e. the first step is to calculate the L.P. diameter

Table of the actual Reduced Mean Pressure for Triples.

Ship's name		Naval Engines									Mercantile Engines															Remarks
		"Wacht"	"Bussard"	"Hohenzollern"	"Gefion"	"Pelikan"	"Kaiserin Augusta"	"Frithjof"	"Meteor"	"Wörth"	"Bayern"	"Ascania"	"Lahn"	"Augusta Victoria"	"Sumatra"	"Scandia"	"Kaiser Wilhelm II."	"Hohenzollern"	"Spree"	"Columbia"	"Fürst Bismarck"	"Normannia"	"Virginia"	"Salier"	"Venetia"	
1		2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
Working pressure in atmos.		10	12	12	12	12	12	12	12	12	10	10	10.5	10.5	11	11	11	11	11	11	11	11.25	11.5	12	12	The engines in Columns 13, 14, 19 to 22 are those of mail steamers. The engines in columns 18 & 24 have 2 H.P. cylinders. The engines in columns 13 & 19 have 2 H.P. and 2 L.P. cylinders.
Cylinder-ratio	H.P.	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	
	M.P.	2.10	2.82	2.61	2.56	2.82	2.48	2.39	2.23	2.49	2.54	2.53	2.19	2.62	2.46	2.58	2.62	2.53	2.00	2.59	2.39	2.80	2.51	2.53	2.58	
	L.P.	4.78	6.97	6.67	5.76	6.97	6.39	5.88	5.88	6.25	6.39	6.95	6.84	6.62	6.67	6.55	6.67	6.77	6.92	6.07	6.00	7.02	6.84	6.72	6.55	
Actual reduced mean pressure in atmos.		2.35	2.43	2.43	2.43	2.45	2.49	2.63	2.72	2.86	2.12	2.08	2.10	2.29	1.98	2.25	2.25	1.97	2.31	2.17	2.20	1.60	2.17	2.10	2.29	

Cylinder-ratio: M.P. $= \frac{\text{M.P.}}{\text{H.P.}}$ cylinder; L.P. $= \frac{\text{L.P.}}{\text{H.P.}}$ cylinder.

by § 49, 1, as if this engine did the whole of the work. This diameter being fixed on, the diameters of the other cylinders are determined, having regard to the points to be observed as detailed in 3).

Actual reduced mean pressure.

2) The actual reduced mean pressures in the table on the preceding page as calculated from the trial-trip cards of a number of naval and mercantile engines will be of assistance in getting out the diameter of the L.P. cylinder. The average values in this table shew that with the usual boiler pressures of 10 to 12 atmos., the reduced mean pressure may be taken at 2 to 2.2 atmos. in mercantile engines and 2.4 to 2.6 atmos. in naval engines. For quadruples of more than 15 atmos. working pressure the reduced mean pressure may be assumed equal to about 3 atmos. The calculation of the reduced mean pressure is explained in § 29, 18, 23 and 24.

Points to be observed in the design of multiple expansion engines.

3) According to the experience hitherto gained it appears to be the most advantageous to have the range of temperature as nearly as possible the same in all the cylinders of a multiple-expansion engine, after which it becomes the more efficient the closer the initial loads and the horse-powers in all the cylinders approach equality. The two latter points however are of comparatively little importance as they only affect the strength of the rods and uniformity of working, but not the economy of the engine. It is impossible to get these three sets of quantities perfectly equal in all the cylinders. If the design is arranged so as to get the initial loads on the pistons equal, the cylinder-ratios thus obtained will differ from those which give an equal distribution of work in the cylinders; so that we are forced to adopt something between the two. If this is carefully done however, it is possible in a triple at any rate, to very nearly fulfil all three requirements. The equality between the quantities referred to is particularly influenced by

1) the sequence of cranks,
2) the distribution of the steam, and
3) the cylinder-ratio.

Crank-arrangement.

4) **II. The Sequence of Cranks** depends primarily upon the arrangement of crank angles. The most usual one for three-crank triples at present is that of dividing the circle into three equal parts. The exceptions to this were described in § 46, 9—12 with their advantages and disadvantages. For quadruples the four-crank arrangement (§ 47, 13), probably on SCHLICK'S plan, will be the favourite. In both cases, — whether 3 or 4 cranks are employed, we may distinguish two different sequences of the cranks in a revolution, viz.

a) high, intermediate, low, and
b) high, low, intermediate.

In the first the H.P. is called the *leading crank*, in the latter which may also be written low, intermediate, high, the L.P. crank is said to lead. The following investigations relate exclusively to triples with three cranks at 120^0, except where some special remark to the contrary is made.

5) a. **When the H.P. crank leads** the piston of the cylinder which is exhausting must travel about two-thirds of its stroke before the admission in the next cylinder occurs. In a small receiver therefore the pressure must rise in proportion to the amount of exhaust from the preceding cylinder. When subsequently the slide of the next following engine opens to admission, the pressure in the preceding cylinder cannot sink to any considerable extent lower, because the compression is about to begin in it. But the beginning of the compression on one side of the piston occurs almost simultaneously with the release on the other side of the preceding piston, and the admission in the succeeding cylinder. From these circumstances the following economical advantages accrue. Firstly, the steam remaining in the receiver after every cut-off in the succeeding cylinder is gradually raised by compression to the pressure of the steam subsequently entering the receiver from the preceding cylinder and this steam is thus protected against condensation losses. For just as the compression of the steam in a cylinder up to the steam-chest pressure is not only accompanied by a certain saving of steam, but also reduces the condensation losses, otherwise unavoidable in consequence of the sudden lowering of the temperature of the entering steam, so a corresponding compression must have a similarly good effect upon the receiver steam. Secondly, it must happen that the steam exhausting from one cylinder passes directly into the succeeding one, thus raising the admission pressure because each slide opens to admission somewhat before the release in the preceding cylinder. This accession of fresh steam to the succeeding cylinder occurs at the moment at which the usual condensation during admission would begin in this cylinder, and this condensation is thus to a great extent prevented by the heat in the new steam. Thirdly, this method of distributing the steam is said to produce a very uniform twisting moment. This proposition is demonstrated by MUDD*) as following from the careful and thorough trials of his first four triple-expansion engines. Two

H.P. crank leading.

*) Proceedings of the Institution of Mechanical Engineers 1886. p. 515.

of these engines had the L.P. crank leading, the other two had the H.P. crank leading and these proved themselves to be the better and more economical engines, so that MUDD always adopts this sequence of cranks. But, as he expressly states, he is obliged to have larger receivers than would otherwise be required and this is the reason why the sequence of cranks he recommends is not more generally adopted, convincing as his arguments are. MORISON*) also defends the large receivers with the H.P. crank leading, for according to his investigations the compression in the receivers extends to about 56.5 % of the stroke (with ordinary valve gear) whereas with the L.P. crank leading, the compression only continues during 19.5 % of the stroke. It therefore happens that in receivers of customary dimensions the compression is too marked, causing a considerable rise of the initial pressure in the M.P. and L.P. cylinders and therefore very heavy loads on the connecting rod brasses which have to be made correspondingly large. Hence it follows that *the H.P. crank leading is perhaps favourable to high economy, but in general requires a heavy engine.*

L.P. crank leading.

6) b. **When the L.P. crank leads** the piston of the smaller H.P. or M.P. cylinder which is exhausting, has only completed about one-third of its stroke when the greater M.P. or L.P. cylinder opens to admission. The contents of their receivers are therefore governed more by the consideration of getting them large enough to supply the succeeding greater cylinder than just small enough to hold the exhaust of the preceding smaller one. With the L.P. crank leading the receivers can therefore be pretty small. But we must not go too far in this direction, for if the receivers are too confined the pressure may rise considerably during the time between the preceding release and the succeeding admission. The larger (within reasonable limits) the receivers are made, the less sensible this compression and its accompanying drawbacks become. This is particularly clearly shewn by the ahead and astern diagrams taken by KENNEDY**) in 1888 during his well-known trial of the triple engine of the S. S. "Meteor". In going ahead the H.P. crank was the leading one and in going astern the L.P. In the latter case the backpressures were much more even than in the former and there was no rise of admission pressure after the beginning of the stroke. *The L.P. crank leading is for the above reasons the better sequence of cranks* and a large portion of the more carefully designed triples have it.

*) The marine engineer. April 1887. p. 5.

**) Proceedings of the Institution of Mechanical Engineers 1889.

7) **III. The distribution of steam** depends upon — Distribution of Steam.
a) the velocity of the steam,
b) the cut-off in the several cylinders.

8) **a. The velocity of the steam** requires attention, in order to prevent the losses of pressure which occur in the cylinders and receivers when the velocity is too great. The lower this is kept, the more closely the actual indicator cards of the engine will agree with the original theoretical ones, as EICKENRODT*) has shewn for a two-cylinder compound engine. The fewer bends, valves, and other constructional details which obstruct the rectilineal flow of the steam are applied and the more carefully the ports, passages, and eduction pipes between the cylinders and into the condenser are designed with a view to easiness, the less wire-drawing there will be at high speeds with its accompanying loss of pressure and effect. These influences of the velocity of the steam are already explained in § 18, 62. At 13 atmos. absolute initial pressure experience has shewn that the velocity of the steam should not much exceed the following. — Velocity of the steam.

Table of Steam-velocities in multiple-expansion Engines.

		Naval Engines	Mercantile Engines
Approximate Revs. per minute		110—120	70—80
Average Piston-speed in m per Sec.		4.0	3.0
H.P. cylinder:	in the Steam-pipe	35 m per Sec.	30 m per Sec.
	„ „ passages	25 „ „ „	20 „ „ „
M.P. cylinder:	„ „ communicating pipe	40 „ „ „	30 „ „ „
	„ „ passages	35 „ „ „	25 „ „ „
L.P. cylinder:	„ „ communicating pipe	45 „ „ „	35 „ „ „
	„ „ passages	40 „ „ „	30 „ „ „

If the steam passages are very indirect, these velocities must be correspondingly reduced, but they may without any great disadvantage be somewhat increased when the passages are particularly easy. The same values can be used for quadruples up to 14 atmos. working pressure.

9) **b. The cut-offs in the several cylinders** are arranged so that either — Cut-offs.
α) the initial loads are equal, or
β) the drop of pressure between the cylinders is avoided; the former distribution is the more usual in marine practice.

10) α. *Equal initial loads* on the pistons mean equal maximum loads on the rods, and as these are always made interchangeable in marine engines it is desirable to arrange the cut-offs so as to get equal stresses on the rods as far as possible. For triples with the usual initial pressures of 11 to 13 atmos. abs., — Cut-offs for equal initial loads.

*) Zeitschrift des Vereins deutscher Ingenieure 1886. p. 215.

cylinder-ratios approximately equal to those given in 18), and receivers of ordinary proportions, the initial loads will most nearly approach equality when the cut-off ratios are 0.5 to 0.6 for the H.P., 0.55 to 0.65 for the M.P., and 0.6 for the L.P., the lower values corresponding to the L.P. engine leading and the higher to the H.P. leading. For quadruples up to 15 atmos. absolute the cut-offs of Nos. II and III must be between 0.55 and 0.65, those of the other engines remain as above.

Cut-offs with no drop.

11) β. *To prevent any drop* the M.P. and L.P. cut-offs must be made to depend in the first place upon the sequence of cranks and, neglecting clearances, then only on the ratios of the cylinders and receivers. As the H.P. and M.P. cranks contain the same angle between them as the M.P. and L.P., the cut-offs for the two latter are got from the same formula. These cut-offs are worked out below for the M.P. cylinder, the volume of which is here called V.

Cut-offs with H.P. crank leading.

12) If the H.P. crank leads and the cylinder ratio $\frac{v}{V} > 0.25$, as is always the case with marine engines, and further, if there is to be no drop between the H.P. and M.P. so that

$$p_\mu \varepsilon_m V = p \varepsilon v, \quad \ldots\ldots\ldots\ldots (271)$$

then by Figs. 1 and 2, Pl. 17, the following equation must be fulfilled,

$$p_\mu \left[\varepsilon_m V + R + (1 - m) v\right] = p\varepsilon \left[0.5\,(1 + \cos 120^0)\, V + R + v\right]$$

$$p_\mu = \frac{p\,\varepsilon\left[0.5\,(1 + \cos 120^0)\, V + R + v\right]}{\varepsilon_m V + R + (1 - m)\, v}$$

Substituting this value in Eq. 271, we get

$$\varepsilon_m = \frac{\frac{v}{V}\frac{R}{V} + (1 - m)\left(\frac{v}{V}\right)^2}{0.25 + \frac{R}{V}} \quad \ldots\ldots\ldots (272)$$

Further if β is the angle between the M.P. crank and the vertical at the moment of M.P. cut-off, then

$$\left.\begin{aligned} \varepsilon_m &= 0.5\,(1 - \cos\beta) \\ m &= 0.5\left[1 + \cos(60^0 - \beta)\right] \end{aligned}\right\} \quad \ldots\ldots (273)$$

Assuming next that $\frac{v}{V} = \frac{R}{V}$, i. e. that the volumes of the first receiver and the H.P. cylinder, and also those of the second receiver and the M.P. cylinder are equal, it follows that

$$\frac{\varepsilon_m}{2 - m} = \frac{\left(\frac{v}{V}\right)^2}{0.25 + \frac{v}{V}} = \frac{1 - \cos\beta}{3 + \cos(60^0 - \beta)} \quad \ldots\ldots (274)$$

13) If the L.P. crank leads and if, as is also invariably the case in marine engines, $\frac{v}{V} < 0.75$ and if further, the conditions of Eq. 271 are complied with, we must have, by Figs. 3 and 4, Pl. 17 Cut-offs with the L.P. crank leading.

$$p_\mu \left[R + (1 - m)\, v\right] = p\, \varepsilon\, R$$

$$p_\mu = \frac{p\, \varepsilon\, R}{R + (1 - m) v}.$$

Substituting this value in Eq. 271, we get

$$\varepsilon_m = \frac{\frac{v}{V}\frac{R}{V} + (1 - m)\left(\frac{v}{V}\right)^2}{\frac{R}{V}} \quad \ldots\ldots\ldots\ldots (275)$$

With the above signification of β

$$\left.\begin{aligned} \varepsilon_m &= 0.5\,(1 - \cos\beta) \\ m &= 0.5\left[1 + \cos(120^0 - \beta)\right] \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots (276)$$

and again if $\frac{v}{V}$ is to be $= \frac{R}{V}$,

$$\frac{\varepsilon_m}{2 - m} = \frac{v}{V} = \frac{1 - \cos\beta}{3 - \cos\,(120^0 - \beta)} \quad \ldots\ldots\ldots\ldots (277)$$

14) From these formulæ HRABAK*) computed a table of which only those values which are applicable to marine engines will be given here. In general those cut-off ratios which will prevent drop do not differ widely from the cylinder-ratio, as was shewn for compounds in § 50, 30. The two ratios would be identical if the receiver volumes were infinite, and they become the more nearly equal the larger the receivers are made. Table.

Cut-off-ratios for M.P. and L.P. Cylinders of Triples with no Drop.

Cylinder-ratio $\frac{V}{v}$		4.00	3.33	2.86	2.50	2.22	2.00	1.81	1.66	1.53
Cut-off-ratio	**H.P. leading**	0.25	0.33	0.40	0.48	0.55	0.62	0.68	0.74	0.80
	L.P. leading	0.30	0.35	0.40	0.44	0.48	0.53	0.57	0.61	0.66

15) It may be seen from the foregoing that with the L.P. crank leading the supply of steam to the M.P. and L.P. cylinders takes place in a normal manner, but that when the H.P. crank Diagrams for comparison.

*) J. HRABAK, Hilfsbuch für Dampfmaschinen-Techniker. Edit. II. Berlin 1891. Part. II. p. 119.

leads these cylinders experience a rise of pressure some time after the admission has begun. HRABAK shews as in Figs. 9 and 10, Pl. 17, how altering the sequence of cranks influences the distribution of the steam and the variation of pressure. These figures are intended to produce equal work in all cylinders, the cylinder-ratio for H.P. crank leading being 1 : 2.92 : 5.55, and for L.P. crank leading 1 : 4 : 6.66; the volumes of first receiver and H.P. cylinder and those of second receiver and M.P. cylinder are equal. The full lines exhibit the simultaneous variations (during admission) of back pressure in the H.P. and M.P. as well as forward pressure in the M.P. and L.P. respectively. The numerals affixed to the most important points of the pressure lines are the absolute pressures in atmospheres and from them the corresponding positions of the several pistons can easily be made out. In Fig. 9 the rise of pressure after the beginning of the stroke is plainly distinguishable both in the M.P. and L.P., the H.P. and M.P. back-pressure line (drawn full) being in two parts.

Determination of the Cylinder-ratio.

16) **IV. The Cylinder-ratio** may be fixed upon with a view to

a) equal ranges of temperature in all the cylinders,
b) „ work „ „ „ „ , or
c) the greatest possible uniformity of running.

The first plan is often adopted in designing a marine engine because it leads most rapidly to provisional dimensions for the cylinders which can afterwards be conveniently modified to suit other requirements. The following reasoning assumes infinitely large receivers, three cranks at 120°, and no rise of pressure after the beginning of the stroke.

Cylinder-ratio for equal ranges of temperature.

17) a. **Equal Ranges of Temperature.** If equal ranges of temperature are exclusively regarded, the ratios of the several cylinders to each other depend upon the initial and terminal temperatures of the steam. At present 11 to 13 atmos. must be regarded as the most usual absolute boiler-pressure for triples, giving an absolute initial pressure in the H.P. of about 10.5 to 12.5 atmos., corresponding to from 181° to 189° C. temperature. At the L.P. release the temperature of the steam is usually about 65° to 80° C. and its pressure from 0.25 to 0.5 atmos. abs. Assuming the temperature to be 70° C., the total range of temperature in the engine will be $181^0 - 70^0 = 111^0$ C., i. e. for each cylinder $\frac{111}{3} = 37^0$ C. By the steam-table on pp. 28 to 31 every kg of steam must therefore have at admission in the respective cylinders the following characteristics.

Triple Expansion Engine	Temperature in °C.	Range of Temperature in °C.	Absolute Pressure in atmos.	Volume in cbm	Temperature in °C.	Range of Temperature in °C.	Absolute Pressure in atmos.	Volume in cbm	Temperature in °C.	Range of Temperature in °C.	Absolute Pressure in atmos.	Volume in cbm	Quadruple Expansion Engine
1	2	3	4	5	6	7	8	9	10	11	12	13	14
Cylinders	10 atmos. abs. press.				13 atmos. abs. press.				19 atmos. abs. press.				Cylinders
H.P. Cylinder	181	37	10.5	0.187	189	40	12.50	0.159	207	34	18.50	0.113	H.P. Cylinder
M.P. Cylinder	144	37	4.1	0 450	149	40	4.75	0.394	173	34	8.75	0.222	I M.P. Cylinder
L.P. Cylinder	107	37	1 4	1.250	109	40	1.45	1 200	139	34	3.60	0.506	II M P. Cylinder
Condenser	70		0 32	5 000	69		0 32	5 000	105	35	1.25	1.380	L.P. Cylinder
									70		0.32	5.000	Condenser

Accordingly the cylinder volumes or, with equal strokes, as invariably in marine engines, the squares of the cylinder diameters will be in the ratio

for 11 atmos. abs. W.P. as $187 : 450 : 1250 = 1 : 2.40 : 6.68$ and

" 13 " " " as $159 : 394 : 1200 = 1 : 2.48 : 7.55$.

Whence the ratios of the diameters is

$$1 : 1^1/_2 \textit{ to } 1^3/_5 : 2^1/_2 \textit{ to } 2^3/_4$$

or approximately in round numbers

$$3 : 5 : 8$$

The columns on the right hand of the preceding table contain the cylinder ratio for the quadruples of the future, viz. for 19 atmos. abs. W.P.

$$113 : 222 : 500 : 1380,$$

or in round numbers $1 : 2 : 4.5 : 12$

and the ratios of the diameters

$$1 : 1.5 : 2 : 3.5 \text{ or } 2 : 3 : 4 : 7.$$

18) In practice the volume of the M.P. cylinder of mercantile engines is made rather larger than 2.4 to 2.5 times that of the H.P. cylinder; for reasons given in 20) it is on an average 2.6 times as large, as shewn in the table on p. 402, whereas the L.P. is mostly below the above proportion or generally only about 6.75 times the H.P. The smaller L.P. cylinder not only reduces the weight and with it the cost, but also the effect of momentum of piston and rod which is very considerable in large engines. The smallest L.P. cylinders or in other words, the largest H.P. cylinders are fitted in naval engines, as the table in 2) shews, in order to get, when required, a higher mean pressure and a considerably increased power out of the engine. The engines of recent war-ships, steaming under normal conditions about 10 to 12 knots and developing, accordingly as they are fitted for natural or forced draught, only $^1/_5$ to $^1/_{10}$ of their maximum power, at which their speed is 18 to 20 knots

Practical Cylinder-ratio.

and above, cannot be in any way compared with mercantile engines, on account of their great reserve of power, which means that economy under ordinary circumstances must be abandoned. For most mercantile engines only produce a speed of 8 to 11 knots when exerting their full power and are expected to work with the greatest possible economy. The conditions of mail steamers and particularly of fast passenger steamers differ again, as they are always steaming at full power in order to obtain before everything, the highest possible speed, and in their case economy can only be a secondary consideration because of the prevailing objectionable competition. Their L.P. cylinders are therefore usually not more than 6 times as large as the H.P. In future mercantile quadruples also it is probable that the L.P. will not be made so large as theoretically necessary, say at most ten times the H.P. Most engines of this kind hitherto built have a cylinder-ratio of 1 : 2 : 4 : 8 with 15 atmos. abs. W.P. The American torpedo-boat "Cushing" (see § 47, 18) has a ratio 1 : 2.04 : 4.03 : 8.17 with 18 atmos. abs. W.P. These are likely to be about the proportions of future naval quadruples.

Cylinder-ratios for equal power.

19) b. **Equal power** in all three cylinders, by Eq. 127, p. 253, requires the fulfilment of this equation

$$\frac{IHP}{3} = \frac{1}{3} D_N^2 p_{i_r} = D_H^2 p_{i_H} = D_M^2 p_{i_M} = D_N^2 p_{i_N}$$

where p_{i_r} is the indicated mean pressure reduced to L.P. piston and p_{i_H}, p_{i_M}, p_{i_N}, the actual indicated mean pressures in the H.P., M.P., and L.P. cylinders respectively. Calling the cylinder-ratios (with equal strokes of course)

$$\frac{D_H^2}{D_N^2} = \varphi \text{ and } \frac{D_M^2}{D_N^2} = \varphi', \text{ we may write}$$

$$\varphi\, p_{i_H} = \varphi'\, p_{i_M} = p_{i_N} = {}^1/_3\, p_{i_r}$$

If p_e is the release pressure in the L.P., then

$$\varphi\, p_{i_H} = p_e \log nat \frac{\varphi}{\varepsilon_g} \text{ and}$$

$$\frac{1}{3} p_{i_r} = \frac{1}{3}\left[p_e\left(1 + \log nat \frac{1}{\varepsilon_g}\right) - a\right]$$

$$p_e \log nat \frac{\varphi}{\varepsilon_g} = \frac{1}{3}\left[p_e\left(1 + \log nat \frac{1}{\varepsilon_g}\right) - a\right]$$

$$\log nat\, \varphi = \frac{1}{3}\left(1 - \frac{a}{p_e}\right) + {}^2/_3 \log nat\, \varepsilon_g \quad \ldots\ldots \quad (278)$$

The value of φ being found from this equation, φ' follows from

$$\varphi\, p_{i_H} = \varphi'\, p_{i_M}$$

which may be expressed thus

$$p_e \log nat \frac{\varphi}{\varepsilon_g} = p_e \log nat \frac{\varphi'}{\varphi},$$

whence $$\varphi' = \frac{\varphi^2}{\varepsilon_g} \quad \ldots\ldots\ldots\ldots\ldots \quad (279)$$

20) As we have been regarding the receivers as infinitely great, whereas in marine practice they are always much contracted, we must now take account of the uneven alteration in the receiver pressures thus caused. Other things being equal, these pressures vary in a manner which differs according to the sequence of cranks, so that we get different cylinder-ratios for the two sequences. A table of these has been worked out by HRABAK*), an extract from which is given below, based on the usual L.P. release pressure in marine engines $p_e = 0.5$ atmos., condenser back-pressure $a = 0.2$ atmos., and the further assumption that the volume of the first receiver equals that of the H.P. cylinder, and that of the second receiver that of the M.P. cylinder. Table.

Cylinder-ratios for Triples with Equal Power in all Cylinders.

Sequence of Cranks		H.P. leading			L.P. leading		
Initial pressure above atmosphere		10 atmos.	12 atmos.	14 atmos.	10 atmos.	12 atmos.	14 atmos.
Cylinder-ratio	$\frac{\text{L.P.}}{\text{H.P.}} = \frac{1}{\varphi}$	5.26	5.55	6.66	5.88	6.66	7.14
	$\frac{\text{L.P.}}{\text{M.P.}} = \frac{1}{\varphi'}$	1.81	1.88	1.96	1.54	1.66	1.75
	$\frac{\text{M.P.}}{\text{H.P.}} = \frac{\varphi'}{\varphi}$	2.86	3.00	3.12	3.84	4.00	4.16
Approximate Cut-off-ratios	Total $= \varepsilon_g$	0.050	0.042	0.036	0.050	0.042	0.036
	H.P. Cylinder $= \varepsilon$	0.26	0.24	0.22	0.30	0.28	0.26
	M.P. Cylinder $= \varepsilon_m$	0.40	0.38	0.35	0.31	0.30	0.29
	L.P. Cylinder $= \varepsilon_n$	0.68	0.65	0.63	0.66		0.59

21) The table shews that for equal powers the M.P. cylinder must be made very large, especially with the usual sequence of the L.P. crank leading. In order to save cost and especially weight, the L.P. cylinders of marine engines are often made *smaller* than given here as the table in 2) shews. If, however an equal distribution of power among the cylinders is desired, this can only be done by allowing a drop to occur Large M.P. cylinder.

*) J. HRABAK. Hilfsbuch für Dampfmaschinen-Techniker. Edit. II. Berlin 1891. II. Part. p. 128.

between the M.P. and L.P. which is the more considerable, the smaller the M.P. cylinder is made. If the H.P. is the leading crank, the M.P. can be made smaller, but then both it and the L.P. experience a rise of pressure after the beginning of the stroke. — For quadruples with equal powers in all the cylinders corresponding cylinder-ratios φ, φ', and φ'' can be evolved without difficulty from 19).

Cylinder-ratios for uniform running.

22) c. **Uniformity of Running**, according to HRABAK, can be obtained to a certain degree in cases where equality of powers is less regarded, by choosing the cylinder-ratios so that the total quantities of work transmitted to the cranks during equal portions of a revolution are equal. For three-crank engines it is handiest to divide the revolution into six parts. If here we again assume the receivers to be infinitely large, the influence of the piston-speed to be neglected, and the steam to be expanded down to the condenser pressure, the required conditions can be fulfilled for either crank leading, by making the cylinder-ratios (of course for equal strokes)

$$\frac{D_N^2}{D_M^2} = \frac{D_M^2}{D_H^2} = \frac{D_H^2}{D_N^2} \times \frac{1}{\varepsilon_g}$$

Cylinder-ratios for uniform running and equal powers.

23) If the steam is expanded down to the condenser pressure, i. e. if $p_e = a$ and the cylinder-ratios are in accordance with the above expression which may be more briefly written

$$\frac{1}{\varphi'} = \frac{\varphi'}{\varphi} = \frac{\varphi}{\varepsilon_g} \quad \dots\dots\dots\dots\dots \quad (280)$$

or if

$$\varphi' = \sqrt[3]{\varepsilon_g},$$

then we must have

$$\varepsilon_g = \varphi\,\varphi' = \varphi'^2$$

$$\varepsilon = \frac{\varepsilon_g}{\varphi} = \varphi'$$

$$\varepsilon_m = \frac{\varphi}{\varphi'} = \frac{\varphi'^2}{\varphi} = \varphi'$$

$$\varepsilon_m = \varphi$$

i. e., the cut-off ratios of all the cylinders are the same, or

$$\varphi' = \frac{D_M^2}{D_N^2} = \sqrt[3]{\varepsilon_g} \quad \dots\dots\dots\dots \quad (281)$$

In this case, which is an ideal one, not only are the sums of the work transmitted to the cranks during every sixth of a revolution equal, but the powers in the three cylinders also.

The expressions for these latter as worked out in 19) are

$$\varphi\, p_{i_H} = p_e \log nat \frac{\varphi}{\varepsilon_g}$$

$$\varphi'\, p_{i_M} = p_e \log nat \frac{\varphi'}{\varphi}$$

$$p_{i_N} = p_e \log nat \frac{1}{\varphi'}$$

and by the assumption in Eq. 280 they must be equal.

24) The adoption of the cylinder-ratios of Eq. 280 will never give equal powers in the three engines in practice, because, for the reason given in § 28, 3, the steam cannot be expanded down to the condenser pressure and further because of the influence of the confined receiver spaces upon the supply of steam to the cylinders. Uniformity of running will be just as imperfectly attained by help of this formula (Eq. 280), because it neglects the influence of momentum. However, the following table of HRABAK'S gives the cylinder-ratios for approximately equal work in each sextant of a revolution for the respective values of ε_g, the total cut-off ratio. **Table.**

Cylinder-ratios (for triples) giving equal Work in each Sextant of a Revolution.

Total Cut-off ratio ε_g	0.100	0.080	0.070	0.060	0.050	0.040	0.030	0.020
$\frac{\text{L.P.}}{\text{H.P.}}\ \frac{1}{\varphi}$	5.00	5.26	5.88	6.66	7.14	8.33	10.31	13.70
$\frac{\text{L.P.}}{\text{M.P.}}\ \frac{1}{\varphi'}$	2.17	2.32	2.44	2.56	2.70	2.94	3.22	3.70
$\frac{\text{M.P.}}{\text{H.P.}}\ \frac{\varphi'}{\varphi}$	2.17	2.32	2.44	2.56	2.70	2.94	3.22	3.70

25) The first five columns of this table contain values which agree very well with the cylinder-ratios of actual engines given in 2). **Average values of cylinder-ratios.** The figures of the last three columns are however not applicable to marine engines, as the cylinders they produce are too large. If the ratios suitable for marine engines are selected from the above table, and it is desired to avoid any drop, an unequal distribution of power over the three cylinders will result. But if this latter quality is objectionable, then either we must put up with the drop, or take an average of the cylinder-ratios given in 20) and 24) which will to a certain extent fulfil both the requirements of equal powers in all cylinders as well as for every sextant of a revolution. — For quadruples with SCHLICK'S crank-angles it is not possible to secure this latter quality by the choice of any particular cylinder-ratio. In these

engines the cylinder-ratios must approach a mean between those for equal ranges of temperature and equal powers.

Advantages of drop.

26) **Drop.** The opinion is very common that a drop of pressure between the cylinders of multiple-expansion engines must be avoided if no loss of effect during expansion is to occur. The unavoidable losses attending the passage of the steam between the cylinders of compounds are noticed in § 49, 15 and a method of minimizing them given. ENSRUD*) has shewn both graphically and analytically that under ordinary practical conditions and neglecting clearances, a small gain of effect accompanies drop provided the cut-off in the succeeding cylinder is properly arranged. If, in a two-cylinder compound the pressure falls from 15 to 20% between the H.P. release and the L.P. admission, a gain of 0.78% of the total power will be obtained by making the L.P. cut-off at 0.36 to 0.38 of the stroke. In the present state of the theory of steam-engines, it is unfortunately impossible to determine either the most favourable amount of drop or the economy resulting from it. It has been said that the gain is so small as to be inappreciable. Even if this is admitted, there still remains the advantage due to the evaporation of the watery particles in the succeeding cylinder by the sudden relief of the exhaust pressure during the drop. Besides, the preceding cylinder can be made smaller for the same range of temperature when drop is allowed to occur, whereby the steam-losses which are a function of the cylinder diameter are lessened. It follows from the above *that under certain circumstances drop influences the economy of the engine favourably instead of unfavourably.* The author**) has examined the indicator cards of a large number of mercantile triples, most of which exhibit very considerable drop, owing to the H.P. and M.P. cylinders being too small. This usually amounts to 30 or 40% on leaving the H.P. cylinder and 20 or 30% on leaving the M.P. It thus appears that a drop of 12% with unjacketed receivers, as here assumed by HRABAK, does not often occur in marine engines, it is generally greater.

Piston-speed.

27) **The Piston-speed of Multiple-expansion Engines** depends upon the dimensions of the L.P. cylinder. The greater its diameter, the heavier of course must be the piston and rods and their acceleration-load. It is generally desirable that the acceleration-load should not exceed the initial steam-load, so that the change of pressure may take place at the dead point and not at some

*) Zeitschrift des Vereins deutscher Ingenieure. 1889. p. 1241.

**) HAACK und BUSLEY. Die technische Entwickelung des Nordd. Lloyd u. s. w. Berlin 1893. Diagrams V to X.

distance from it, as this is considered to cause an uneven motion in the engine (compare 39). But as the L.P. engine always has the greatest weight of rods combined with the smallest initial pressure, the piston-speed of a triple cannot, according to RADINGER*), exceed

$$c = \sqrt{\frac{p - a}{\frac{O}{P}\left(1 + \frac{r}{l}\right)}} \text{ metres.}$$

The initial pressure $p - a$ in the L.P. cylinder of most mercantile engines is actually not more than 0.8 atmos., in naval engines it is usually 1 atmos., reaching 1.6 atmos. and above only in particularly successful forced-draught trials of such engines especially in those of torpedo-boats and catchers. As further, the value of $\frac{O}{P}$ for the L.P. cylinder in mercantile engines is seldom below 0.08 atmos., whereas for the engines of battle ships and torpedo-boats it falls to 0.06 or 0.04 atmos. (see § 48, 13), the following approximate highest, mean, and lowest piston-speeds come out for a length of connecting rod $l = 4r$

Actual initial pressure $p - a$ atmos.	1.6	1.0	0.8
Weight of the piston and rods in kg per □cm of piston area	0.04	0.06	0.08
Approximate piston-speed c in m	5.66	3.25	2.83

Accordingly, fast-running engines must always have smaller L.P. cylinders with much lighter pistons and rods than slower ones. A means of increasing the acceleration-pressure of the L.P. piston and rods by means of tandem cylinders and its attendant disadvantages are described in § 46, 13. — It will not be possible to greatly exceed the above limits of piston-speed in quadruples with higher pressures unless a still greater reduction in the weight of the rods &c. can be introduced.

28) **The clearance spaces of multiple-expansion engines,** measured from a number fitted, may be taken as the following fractions of the respective volumes swept; Clearance.

	H.P.	M.P.	L.P.
for ordinary slides	0.08 to 0.12	0.07 to 0.11	0.05 to 0.10
for piston-valves	0.09 „ 0.13	0.10 „ 0.14	0.11 „ 0.15

*) J. RADINGER. Ueber Dampfmaschinen mit hoher Kolbengeschwindigkeit. Edit. III. Vienna 1892. p. 59.

Table of Performances of

Ship's name	Pressure above atmo: in the boilers (atmos.)	Pressure above atmo: at H.P. admission (atmos.)	Pressure above atmo: Loss (atmos.)	Piston-speed (m/sec)	Clearance in % of volume swept: H.P.	Clearance in % of volume swept: M.P.	Clearance in % of volume swept: L.P.	Indicator diagrams: actual expansions	Indicator diagrams: Fulness of the Indicator diagrams	Indicator diagrams: mean indicated pressure in H.P. (atmos.)	Indicator diagrams: mean indicated pressure in M.P. (atmos.)	Indicator diagrams: mean indicated pressure in L.P. (atmos.)	Temperature of the steam: at admission H.P. (°C)	Temperature of the steam: L.P. release (°C)	Range of: Total (°C)	Range of: mean for each Cylinder (°C)
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
Ia. Three-cylinder Engines in																
"Bayern"	10.10 9.80	9.80 9.50	0.30 0.30	3.47 3.25	11	10	12	8.616 8.616	0.550 0.480	3.700 4.200	1.770 1.462	0.840 0.565	183 181	72 76	111 105	37 35
"Ascania"	10.40	9.85	0.55	2.60	11	7	6	9.636	0.570	4.500	2.031	0.699	183	75	108	36
"Scandia"	10.90	10.20	0.70	3.17	16	13	11	9.652	0.568	4.950	1.850	0.770	184	64	120	40
"Sumatra"	10.80	10.30	0.50	1.97	10	8	5	13.125	0.590	5.450	1.700	0.471	184	58	126	42
"Virginia"	11.30	10.80	0.50	2.92	top 16.0 bot 12.8	top 10.6 bot 10.6	top 5.9 bot 6.0	9.380	0.590	5.125	1.847	0.745	186	63	123	[illegible]
"Venetia"	12.00	11.25	0.75	2.59	10	7	5	9.545	0.590	4.750	2.025	0.780	188	68	120	[illegible]
"Kaiser Wilhelm II."	10.80	9.80	1.00	3.73	top 16.6 bot 17.8	top 12.4 bot 13.4	top 16.5 bot 16.6	8.230	0.580	4.700	1.940	0.773	183	69	114	[illegible]
Ib. Three-cylinder Engines																
"Augusta Victoria"	10.50	9.50 9.80	1.00 0.70	4.05	top 16.6 bot 17.8	top 12.4 bot 13.4	top 16.5 bot 16.6	7.607 7.880	0.556 0.580	4.440 4.450	1.830 1.980	0.815 0.840	181 183	67 66	114 117	[illegible] 39
"Fürst Bismarck"	10.60	10.00 10.20	0.60 0.40	4.87 4.82	15	13	13	8.071 8.370	0.598 0.574	4.000 4.120	1.650 1.850	0.830 0.840	183 184	66 67	117 117	[illegible] [illegible]
"Columbia"	10.40	9.84 9.98	0.56 0.42	4.23 4.13	10	14	18	8.640 8.307	0.574 0.560	4.530 4.420	1.620 1.690	0.724 0.731	182 183	74 72	108 111	[illegible] [illegible]
"Normannia"	11.11	9.91 9.50	1.20 1.61	4.86	16.6	14.1	15.6	12.888 11.318	0.543 0.560	3.670 4.220	1.130 1.110	0.590 0.580	183 181	66 64	117 117	[illegible] [illegible]
II. Four-cylinder Engines																
"Hohenzollern"	11.0	10.45	0.55	2.94	top 9.0 bot 16.0	9.5	7.3	12.706	0.594	f. 4.900 a. 4.750	1.620	0.650	185	59	126	[illegible]
"Salier"	12.0	11.50	0.50	3.00	top 7.0 bot 15.0	10.0	12.4	16.143	0.619	f. 4.900 a. 4.450	1.840	0.710	189	60	129	[illegible]
III. Five-cylinder Engines																
"Lahn"	10.5	9.35	1.15	4.27	20.0	10.7	10.9	10.909	0.560	f. 4.080 a. 4.400	2.470	f. 0.680 a. 0.700	180	60	120	[illegible]
"Spree"	11.0	10.50	0.50	4.22	19.4	14.7	12.3	11.200	0.570	f. 4.600 a. 4.350	3.150	f. 0.740 a. 0.770	185	62	123	[illegible]

[1]) Initial load of the forward pair of Cylinders: 52138 kg;
[2]) " " " " " " " " 42151 kg;
[3]) " " " " " " " " 58119 kg;
[4]) " " " " " " " " 86703 kg;

19 Triple-expansion Engines.

Temperature actual in H.P. °C	Temperature actual in M.P. °C	Temperature actual in L.P. °C	Initial load in H.P. kg	Initial load in M.P. kg	Initial load in L.P. kg	Horse-power Total IHP	Horse-power in H.P. IHP	Horse-power in M.P. IHP	Horse-power in L.P. IHP	Horse-power per □m of Grate IHP	Coal-consumption per IHP per hour kg	Coal-consumption per □m of grate per hour kg	Sequence of Cranks	Remarks
18	19	20	21	22	23	24	25	26	27	28	29	30	31	32
in Mail and Cargo-steamers.														
34	37	39	39024	51190	49858	4088	1114	1361	1613	122.3	0.695	84.9	H. M. L.	Trial-trip
31	35	27	37073	43760	33238	3247	1184	1046	1017	97.1	0.800	77.6	H. M. L.	Voyage
37	34	38	15902	18373	21470	1193	379	405	409	118.9	0.780	92.7	H. M. L.	"
37	40	43	37685	45088	42476	3463	1160	1120	1183	118.2	0.729	86.2	H. M. L.	Trial-trip
46	43	38	8144	6618	4755	335	141	110	84	98.5	0.750	74.3	H. L. M.	"
43	43	45	25093	26941	26443	1836	624	576	636	107.0	0.700	74.9	H. M. L.	"
39	41	43	24339	32300	29087	1678	522	584	572	118.2	0.690	81.6	H. L. M.	"
40	40	40	58881	68094	62980	6511	2053	2223	2235	119.0	0.781	92.9	H. M. L.	Voyage
in "fleet" Steamers.														
37	38	42	55418	69910	65557	6889	2091	2260	2538	135.0	0.871	117.6	H. M. L.	Stbd.-Engine Voyage
38	38	42	57149	71272	64412	7157	2096	2445	2616					Port "
40	43	47	66521	78308	79012	8047	2488	2450	3109	121.5	0.857	104.1	H. M. L.	Stbd. " Trial-trip
40	44	47	62871	83983	80729	8364	2534	2718	3112					Port "
37	30	34	54963	57356	51700	6116	2082	1962	2072	116.1	0.981	104.6	H. L. M.	Stbd. " Voyage
37	34	36	56152	61106	49890	6034	1985	2007	2042					Port "
43	36	38	59503	59020	53950	6234	2083	1798	2352	94.1	0.862	81.1	H. L. M.	Stbd. " Trial-trip
46	30	37	58364	44719	47925	6472	2395	1766	2311					Port "
in Mail-steamers.														
f. 42 a. 47	40	43	f. 18472 a. 20197	33666 [1]	31995	2570	f. 472 a. 458	791	849	133.8	0.697	92.2	—	Trial-trip
f. 39 a. 43	42	42	f. 13752 a. 15019	28399 [2]	23598	2042	f. 355 a. 322	674	691	170.1	0.696	118.4	—	"
in "fleet" Steamers.														
f. 33 a. 37	40	f. 39 a. 42	f. 31006 a. 36352	72850 [3]	f. 27113 a. 30777	8761	f. 1242 a. 1338	3306	f. 1417 a. 1458	106.0	0.739	80.0	—	Trial-trip
f. 35 a. 33	46	f. 40 a. 40	f. 48907 a. 43236	116247 [4]	f. 37796 a. 34361	12774	f. 1837 a. 1737	5031	f. 2043 a. 2126	122.8	0.693	82.1	—	"

[1] Initial load of the after pair of Cylinders: 52192 kg.
[2] " " " " " " " " 38617 kg.
[3] " " " " middle " 72850 kg. Initial load of the after pair of Cylinders: 67129 kg.
[4] " " " " " " 116247 kg. " " " " " " " " 77597 kg.

The above figures apply when each cylinder has *one* slide. If the M.P. or the L.P. have each *two* piston-valves, the clearance rises to 0.15 or 0.16, while in the L.P. cylinders of the largest engines with *four* piston-valves it is as high as 0.20. These values are means of top and bottom but it is advisable in vertical engines to make the bottom clearance about 10% greater than the top to allow for the engines working down. For quadruples the clearance of the second M.P. cylinder can be assumed to be about the same as that of the first and equal to that given above for the M.P. of triples.

Fulness. 29) **The fulness of the combined indicator diagrams** of triples constructed by WYLLIE'S method (§ 18, 47) and taking clearance into account, comes out at about 0.55 to 0.60 or say on an average 0.57. When below this they are unsatisfactory and when above it, good or very good. With quadruples the fulness is rather lower, so that 0.58 is an excellent figure for them. — When the cards are combined *without* reference to clearance, the fulnesses quoted in § 29, 23 and 24 are obtained.

Table. 30) The table on pp. 500 and 501 contains the most important of the values found from indicator-cards investigated by the author. Of these the most interesting are the range of temperature, the initial pressure, and the power in the several cylinders. The diameters and ratios of the cylinders as well as the builders' names are in the table on p. 402, and the actual reduced mean pressures in that on p. 485. Most of the cards were taken on trial-trips, a small portion being taken on voyages, as remarked in the table. The cards themselves are published in the work mentioned below*).

Kohn's Diagram. 31) **V. Graphic method of determining the working volume of steam.** The construction of theoretical indicator diagrams for a triple or quadruple is essentially more complicated than for a compound, because the calculation of the volume of steam enclosed at any moment between two of the pistons can only be accomplished with difficulty, — even neglecting clearance, thus rendering the determination of the corresponding pressures a very lengthy affair. SCHRÖTER'S graphic method of computing the volume of steam in two cylinders in communication with each other is described in § 51, 28. This process involves the construction of an ellipse whose axes are inclined to the axes of coordinates of the diagram if the cranks enclose an acute or obtuse angle, as for instance in SCHLICK'S arrangement. In the method pro-

*) HAACK und BUSLEY. Die Entwickelung des Norddeutschen Lloyds u. s. w. Berlin 1893. Diagram-sheets V to X.

posed by KOHN*) now to be described, the drawing of two circles suffices to determine the steam volumes for two cylinders in communication at any position of the cranks. It further exhibits the cut-off ratios of the M.P. and L.P., as soon as that of the H.P., together with the volumes of the cylinders, the clearances, and the receivers are fixed upon. There are thus two problems to solve, viz.

a) determination of steam-volume of communicating cylinders,

b) „ „ M.P. and L.P. cut-off ratios.

32) a. **The determination of the steam-volume** is worked out below for the M.P. cylinder of a triple with cranks at 120°, and the L.P. crank leading. The volume enclosed between the M.P. and L.P. pistons, is obtained by an analogous process, which can of course be modified to suit the case of the H.P. crank leading. In Fig. 5, Pl. 16 let I be the H.P. and II the M.P. cylinder which are in communication through the receiver R, as drawn. If these cylinders with equal strokes are conceived to be replaced by two others of unit sectional area but of the same volumes as I and II respectively and therefore of correspondingly differing strokes (as in a combined diagram), then r_1 and r_2 (Fig. 6, Pl. 16) will be the crank-arms modified accordingly. In the same way the volumes of the clearances of both cylinders and of the receiver are to be replaced by ideal cylinders of unit sectional area and the heights h'_s, h''_s, h_r. At the dead-point of crank r_2 the steam-volume enclosed between the two pistons (Fig. 6, Pl. 16) is Analytical process.

$$V_0 = h'_s + h''_s + h_r + r_1 + c.$$

Produce r_1 through o to 3, draw the line $\overline{31} = m$, then

$$m \cos \mu = r_2 - c$$

therefore $V_0 = h'_s + h''_s + h_r + r_2 + r_1 - m \cos \mu$

or, calling the constant $h'_s + h''_s + h_r + r_1 + r_2 = h$,

$$V_0 = h - m \cos \mu$$

If the crank r_2 revolves through an angle α, the steam-volume in question is increased by a and diminished by b, therefore

$$V_\alpha = h - [m \cos \mu - a + b];$$

but as the expression in the bracket is the horizontal projection of m, we get

$$V_\alpha = h - s' = h - m \cos (\alpha - \mu) \quad \ldots \ldots \ldots (282)$$

We know that the steam volume for a single-cylinder engine is

$$V = a + r (1 - \cos \mu),$$

where a is the height of a cylinder of unit sectional area, the volume of which equals that of the clearance, or

$$V = (a + r) - r \cos \mu \quad \ldots \ldots \ldots \ldots (283)$$

*) Zeitschrift d. Vereins deutscher Ingenieure 1889. p. 1215.

A comparison of equations (282) and (283) shews that the volume of steam enclosed between the pistons of the communicating cylinders varies as if the steam were in one cylinder, of which the stroke is $2\,m$, and the clearance volume equals that of a cylinder of unit sectional area and height $h - m$, while the crank m, the *relative crank*, forms an angle μ with the crank r_2.

Construction of the Diagram. 33) From the above there is evidently no difficulty in representing the steam-volumes corresponding to the various positions of the cranks by means of a polar diagram like ZEUNER'S. If the crank-arm r_2 is laid off from $\overline{0\ 1}$ (Fig. 7, Pl. 16) and afterwards from the point 1, $r_1 = \overline{13}$ is also laid off at the angle β, i. e. the angle included between the crank-arm r_1 and a horizontal line (see Fig. 6), then $\overline{03}$ gives the relative crank m and $\overline{301}$ is the angle μ. Upon m as diameter describe a circle and another with 0 as centre and $h = \overline{04}$ as radius, also draw a radius from 0 at an angle α to the horizontal, then the portion $\overline{65}$ of this radius which lies between the two circles, represents the volume V_α, because $\overline{65} = h - m \cos(\alpha - \mu)$. When the crank r_2 revolves, the portion of r_2 produced which lies within the shaded area between the two circles, represents the steam-volume enclosed between the pistons of the two communicating cylinders for any position of the cranks. It is now seen that this volume is a minimum and therefore the pressure a maximum, when the crank r_2 is at the angle μ from the centre line. The third circle, the diameter of which $\overline{07} = m$ forms with r_1 the angle μ, enables the steam-volumes corresponding to any position of the crank $r_1 = \overline{08}$ to be determined.

Assumptions. 34) b. **The determination of the cut-off ratio in the M.P. cylinder** of the engine referred to in 32) is arrived at in the following manner. The cut-off in a cylinder which is being supplied with steam from another one, should take place as soon as it has received a weight of steam and condensed water equal to that supplied per stroke by the boiler to the engine. On the reasonable assumption that this mixture is uniform throughout the communicating spaces, the weights of mixture contained in them will be proportional to their volumes. We will call

D_s' and D_s'' the weight of mixture in the H.P. and M.P. clearances at the moment when the communicating spaces are completely isolated,

D_R the weight of mixture in the M.P. receiver,

D the steam per stroke supplied to the engine.

Analysis. 35) At the moment of the M.P. cut-off, let the volume swept by the M.P. piston be V_x, the volume enclosed between the two

pistons V, and β the corresponding angle of the crank r_2. By the foregoing we have

$$\frac{V_x}{V} = \frac{D}{D + D'_s + D''_s + D_R}.$$

But as $V_x = r_2\,(1 - \cos\beta)$ and $V = h - s_\beta$,

$$h = s_\beta = C r_2\,(1 - \cos\beta) \quad \ldots\ldots\ldots\ldots \quad (284)$$

where

$$C = \frac{D + D'_s + D''_s + D_R}{D}.$$

By a transformation of Eq. (284) we get

$$s_\beta - C r_2 \cos\beta = h - C r_2 \quad \ldots\ldots\ldots\ldots \quad (285)$$

36) If in the volume-diagram (Fig. 8, Pl. 16), we draw a circle with v as centre and $o\,a = h - C r_2$ as radius and another circle on $o\,b = C r_2$ as diameter, then the radius drawn from o through e the point of intersection of the two circles, gives the position of the crank r_2 at which the M.P. cut-off is to take place, for at this position we have by construction Construction of the diagram.

$$s - C r_2 \cos\beta = e\,f = h - C r_2$$

fulfilling Eq. (285).

37) **VI. Investigation of the uniformity of running.** This subject is discussed in § 48 for a single-expansion engine both with reference to the design and to trials after completion. As an example of the easiest and most perspicuous method for this purpose, the cards*) of *one* engine are here given of the *ci-devant* British mail-steamer "City of Paris" taken on May 5th 1889 during an outward voyage on which**) her average speed was 19.95 knots. The particulars of these engines which concern us here are the following Indicator diagrams.

	H.P. Cylinder	M.P. Cylinder	L.P. Cylinder
Diameter in m	1.143	1.803	2.870
Stroke " "	1.524	1.524	1.524
Weight of piston, tons	1.150	2.450	5.500
" " piston-rod with cross-head and guide-shoes, tons	4.000	4.000	4.000
" " connecting rod "	5.800	5.800	5.800
" " crank "	2.260	2.260	2.260

It is advisable first to construct a combined diagram Fig. 8, Pl. 18, from the indicator cards after they have been averaged, in order to ascertain by its coefficient of fulness the economy of the engine as a steam user. A coefficient of 0.6 is to be regarded as satisfactory.

*) Zeitschrift des Vereins deutscher Ingenieure. 1890. Pl. XXXIX.

**) Engineering 1891. I. p. 725.

Köhler's Vertical-load diagram.

38) From the indicator cards the steam pressure diagram (§ 48, 26) is first to be constructed, then the initial acceleration pressure $\frac{F}{O}$ (§ 48, 14) calculated from the above dimensions and weights, and afterwards the acceleration pressure diagram (§ 48, 29) got out for the finite connecting rod ($l = 4\,r$). The vertical pressure diagrams (§ 48, 18) can then be drawn, KÖHLER'S*) method shewn in Figs. 4 to 6, Pl. 18 being the handiest. The piston strokes are stepped off successively on the X axis, AB representing the down-stroke, BC the up-stroke, &c. The dotted line at the top and bottom of these diagrams exhibits the effect of the weights, sinking during the down-stroke and having to be lifted during the up-stroke.

Reversal of Pressure.

39) It may be seen from the vertical-load diagrams of the three cylinders that the acceleration pressures considerably exceed the steam pressures, so that the change of pressure does not occur at the dead point, as is the case when these pressures are equal, a suitable piston speed having been adopted, — but takes place during the first half of the stroke (compare 27). The greater this excess of acceleration pressure over steam pressure, that is, the higher the piston-speed, the further the point of change of pressure shifts away from the dead point, as is particularly obvious in the L.P. diagram. But engineers are universally of opinion that unless these points coincide the engine will work unevenly and knock, and KÖHLER has demonstrated from his investigations of various engines that so far as the intensity of the blows is concerned there is no material difference whether the reversal of pressure takes place before or after the dead point. In marine engines it is quite likely that the reversal does occasionally occur at some distance from the dead point with the high piston-speeds reached on trial-trips and specially forced passages, though it would be more advisable to maintain the coincidence of these points as has been hitherto considered necessary.

Twisting moment diagrams.

40) From the vertical-pressure diagrams the twisting moment diagrams for the several cylinders can now be easily drawn, and from these the combined twisting moment diagram by § 48, 20 and § 49, 5, as shewn in Fig. 7, Pl. 18. But in order to more clearly exhibit the negative moments, the moments are not set off as in Fig. 6, Pl. 14 from the centre of the circular diagram, but as in Fig. 7, Pl. 18, from the periphery of a circle for which the crank-pin circle may be most convenienty used. Care must be taken in drawing this circular diagram to adhere to

*) Zeitschrift des Vereins deutscher Ingenieure. 1891. p. 83.

the correct positions of the cranks. Their corresponding piston positions with the length of connecting rod $l = 4\,r$ are therefore to be determined on the vertical-pressure diagrams and the respective ordinates converted into tangent loads by § 48, 30. As the indicator cards are always of different scales, the ordinates of the vertical-pressure diagram must first all be brought to the same scale. In the present case they are all reduced to the H.P. scale and therefore correspondingly shortened in the M.P. and L.P. diagrams. The ordinates representing the loads are of course increased in length in the diagrams for these cylinders in the ratios of their piston-areas to the H.P. piston area. According to the particulars given in 37) these ratios for the "City of Paris" are 1 : 2.5 : 6.33 and as the scales of the cards are as 3 : 6 : 16, the ordinates of the M.P. and L.P. diagrams must be increased as $6.33 \times \frac{3}{16} = \frac{19}{16}$ and $2.55 \times \frac{3}{6} = 1.25$, keeping those of the H.P. unaltered. The circle of mean twisting moment, the area of which equals the area of the combined twisting moment diagram (its radius being 1) shews that in the "City of Paris" the ratio of maximum to mean twisting moment is 1.25 to 1. — Instead of the circular diagram here shewn, the extended diagram can be used as represented in Fig. 14, Pl. 17 for the S. S. "Westmoreland" *) the circular diagram for which is given in Fig. 15. According to this her ratio of maximum to mean twisting moment is 1.135 to 1, an extraordinarily favourable one (compare table on p. 435). The "Westmoreland's" engines must therefore work much more evenly than those of the "City of Paris".

*) Engineering. 1886. II. p. 72.

PART II

MARINE BOILERS.

Ninth Division.

Types of Marine Boilers.

§ 53.

Efficiency of Marine Boilers.

1) **I. Definitions.** The excellence of a marine boiler is judged of by its efficiency. We may distinguish the following two significations of the word efficiency according to whether we regard only the communication of heat to the water, or also take into account that the boiler with the engine is concerned in converting as completely as possible the heat stored up in the products of combustion into mechanical work, — viz. **Definitions of efficiency.**

a) the ordinary efficiency according to RANKINE,

b) the thermodynamic efficiency according to LORENZ.

2) The following definitions will be used in the discussion of these two ways of regarding the efficiency; **Further definitions.**

Q_0 the quantity of heat developed from the fuel in the furnace,

Q_1 " " " " given up to the boiler water,

Q_2 " " " " " " by the gases when they are cooled down to the temperature of the external air,

W the weight in kg of gases of combustion leaving the furnace per hour,

F the weight in kg of fuel burnt per □m of grate per hour,

c the specific heat of the gases at constant pressure,

T_0 their temperature of combustion on the grate,

T_1 " " at first contact with the heating surface,

T_2 " " on leaving the boiler.

T_s the temperature of the boiler water,

T_l " " " " external air,

S the heating surface of the boiler in □m,

ds an element of the heating surface,

q the heat given up by the gases to the boiler water per □m of heating surface per hour,

φ the coefficient of transmission of heat,

a, A, and B are coefficients, the signification of which will be given as they occur.

Explanation.

3) **II. Boiler efficiency according to Rankine** *). The ordinary meaning of the efficiency of a boiler is the ratio of the heat Q_1 imparted to the boiler water, to the heat Q_0 generated from the fuel in the furnace, or in other words the ratio of the weight of water actually evaporated per kg. of fuel by the boiler to the weight of water which ought theoretically to be evaporated by 1 kg. of the fuel.

Heat given off by the gases.

4) The quantity of heat transmitted per hour from the gases through an element ds of the heating surface to the water is

$$q\,ds$$

therefore the gases in passing over the element ds experience a diminution of temperature

$$d\,T = \frac{q\,ds}{c\,W}$$

But as the temperature of the gases in their passage over the entire heating surface $S = \int ds$ falls from T_1 to T_2, we get for the total heat transmitted per hour through the entire heating surface

$$\int \frac{ds}{c\,W} = \int_{T_2}^{T_1} \frac{d\,T}{q}$$

The constant a.

5) Experiments have shewn that the coefficient of transmission q for the boiler surfaces is approximately proportional to the square of the difference between the temperatures of the gases and the water

$$q = \frac{(T_1 - T_s)^2}{a}$$

where a is a constant ranging between 20 and 25 per □m of heating surface per hour.

Efficiency of the heating surface.

6) Substituting this value of q in the last Eq., we get

$$\int \frac{ds}{c\,W} = \int_{T_2}^{T_1} \frac{a\,dT}{(T_1 - T_s)^2}$$

$$\frac{S}{c\,W} = a\left(\frac{1}{T_2 - T_s} - \frac{1}{T_1 - T_s}\right),$$

and on bringing the right side to a common denominator,

$$\frac{S}{a\,c\,W} = \frac{T_1 - T_2}{(T_2 - T_s)\;(T_1 - T_s)}.$$

*) W. J. M. Rankine. A manual of the steam engine. pp. 262 & 293. London 1873.

Substituting in this expression the value of $\frac{1}{(T_2 - T_s)}$ obtained from the above Eq., viz

$$\frac{1}{(T_2 - T_s)} = \frac{S}{a\,c\,W} + \frac{1}{T_1 - T_s}$$

we have

$$\frac{S}{a\,c\,W} = \frac{T_1 - T_2}{T_1 - T_s}\left(\frac{S}{a\,c\,W} + \frac{1}{T_1 - T_s}\right)$$

or

$$\frac{T_1 - T_2}{T_1 - T_s} = \frac{S}{a\,c\,W\left(\frac{S}{a\,c\,W} + \frac{1}{T_1 - T_s}\right)} = \frac{S}{S + \frac{a\,c\,W}{T_1 - T_s}}.$$

Again, calling the quantity of heat developed from the gases while their temperature falls from T_1 to that of the water T_s, H, we have further

$$T_1 - T_s = \frac{H}{c\,W}$$

and, substituting this in the last Eq.

$$\frac{T_1 - T_2}{T_1 - T_s} = \frac{S}{S + \frac{a\,c^2\,W^2}{H}} \quad \ldots\ldots\ldots\ldots \quad (286)$$

The quotient $\frac{T_1 - T_2}{T_1 - T_s}$, i. e. the ratio of the heat actually imparted to the water to the heat lost by the gases on their way from the grate to the uptake is called *the efficiency of the heating surface.*

7) The efficiency of the boiler is a function of the efficiency of the heating surface, which may be expressed thus **Coefficient *B*.**

$$\frac{Q_1}{Q_0} = B\,\frac{S}{S + \frac{a\,c^2\,W^2}{H}},$$

where B is a coefficient determined by experiment, representing all the heat losses occurring in the furnace generally (see § 21).

8) Now $c^2\,W^2$ is nearly proportional to $F^2\,L^2$, F being the coal burnt per hour in kg. and L the necessary weight of air to be determined by Eq. 99. Then **Coefficient *A*.**

$$\frac{a\,c^2\,W^2}{H} = \frac{a\,F^2 L^2}{H} = \frac{a\,L^2\,F}{H}\,F = A\,F,$$

so that $\frac{a\,c^2\,W^2}{H}$ is a function of F which can be calculated when the constant A is known. Its value has been ascertained empirically and is approximately proportional to the square of the weight of air required per kg. of fuel.

Efficiency.

9) As it is usual to express the heating surface of a boiler as so much per sq. m of grate, we can call S in the above Eq. *the number of sq. m of heating surface per sq. m of grate* if F is correspondingly understood to represent *the number of kg of coal burnt per sq. m of grate per hour.* The efficiency of the boiler may then be expressed by

$$\eta = \frac{Q_1}{Q_0} = B \frac{S}{S + AF} \quad \ldots\ldots\ldots\ldots \quad (287)$$

Value of the formula.

10) This formula which is only intended to serve as a guide to the designer or for comparing boilers of various systems, cannot of course be expected to agree accurately with the results of experiments and trials which may be instituted with any particular boiler.

Assumptions for the formula.

11) In applying the formula (287) it is assumed that

a) the heating surface of feed-heaters and superheaters is included in S,

b) the heating surface in the boiler is that portion only which is in contact with the water,

c) the air-supply and the grate are so arranged that there can be no noticeable losses of heat due either to imperfect combustion or excess of air,

d) the arrangement of the furnace is adapted to the fuel used.

Rankine's coefficients.

12) The following table of the coefficients A and B is based on the above assumptions.

Table of the Coefficients A and B.

Arrangement of Boiler	A	B
Best convection, natural draught produced by chimney	0.100	1.00
Ordinary convection, chimney draught	0.100	0.92
Best convection, forced draught	0.067	1.00
Ordinary convection, forced draught	0.067	0.95

By the best convection is understood that the feed-water enters the boiler at the coldest part and travels gradually to the hottest part or else that it passes through a feed-heater or an economizer before reaching the boiler.

Wilson's coefficients.

13) WILSON*) gives the same coefficients for RANKINE'S formula except that he puts B at only 0.8 for cylindrical boilers under natural draught and with this he calculates the following

*) R. WILSON. A treatise on steam boilers. London 1879. p. 295.

Table of Efficiency of Boilers under Natural Draught.

Type of Boiler	Heating surface in sq. m per kg of coal burnt per hour	Water evaporated from 100° C. per kg of coal	Efficiency of Boiler
Tubular Boiler	3.5	6.72	0.48
"	5.0	7.42	0.53
"	6.0	7.48	0.56
"	7.0	8.40	0.60
"	8.5	8.68	0.62
"	10.0	8.96	0.64
Water-tube Boiler	15.0	9.66	0.69
"	20.0	9.94	0.71
"	25.0	10.00	0.72
"	30.0	10.20	0.73

Tubular boilers are those in which the flame passes through the tubes and the water surrounds them externally, in water-tube boilers the reverse is the case.

14) **III. Thermodynamic Efficiency of Boilers according to Lorenz*).** Starting with the idea that the boiler with the engine *together* should convert as much as possible of the heat stored up in the products of combustion into mechanical work, we will first seek to determine a factor of conversion for it. We are only able to do this by comparing the process in the boiler and engine with a perfect cycle undergone by a medium (for instance water in the liquid and gaseous states) taking up heat from one body (the furnace-gases), converting part of it into work, and rejecting the balance to another body (the air). Explanation.

15) If the temperature of combustion is regarded as the highest and that of the external air as the lowest in the cycle, or that down to which the gases could theoretically be cooled, then, referring to § 7, 14, the course of such a perfect cycle would be as follows. The working medium (the water) having originally the combustion temperature T_0, expands while its temperature always remains the same as that of the furnace-gases, which falls as they give up their heat. Then a temperature interval of dT corresponds to the transmission of a quantity of heat Perfect cycle.

$$dQ_0 = c\,W\,dT.$$

Calling the simultaneous initial temperature of the gases and of the working medium (the water) T_0 and the terminal temperature of both T_l, the whole heat taken up by the working substance is

$$Q_0 = c\,W(T_0 - T_l).$$

*) Zeitschrift des Vereins deutscher Ingenieure. 1894, p. 1450.

Having arrived at the terminal temperature the body is again condensed, a heat quantity Q_2 being withdrawn from it. We will regard this as being effected by the evaporation of K kg of cooling water per hour at the constant temperature T_l as the calculation is simpler than if the heat is assumed to be given up to an infinitely great heat-receiver. Calling r the latent heat of the steam at T_l, we get

$$- d\,Q_2 = r\,d\,K$$
$$- Q_2 = r\,K.$$

Finally the working medium is to be restored to its initial state corresponding to the temperature T_o, by means of adiabatic compression. The cycle which is reversible in all its parts then begins over again, so that the equation of CLAUSIUS (see § 7)

$$\int \frac{d\,Q}{T} = 0$$

is applicable to it and, being split up into the three periods in question, takes the form

$$\int_{T_l}^{T_o} \frac{d\,Q_o}{T} + \int_{T_l} \frac{d\,Q_2}{T} + \int_{T_o}^{T_l} \frac{d\,Q}{T} = 0.$$

If the above values of Q_0 and Q_2 are substituted in this expression and it is noted that the third summand which refers to the adiabatic compression, viz. $d\,Q = 0$, disappears, also that the temperature and the heat of evaporation both remain constant during the second change of state while heat is being withdrawn, the equation takes the form

$$\int_{T_l}^{T_o} c\frac{W\,d\,T}{T} - \frac{1}{T_l}\int_{o}^{K} r\,d\,K = 0$$

$$c\,W \log nat \frac{T_o}{T_l} = \frac{r\,K}{T_l}.$$

From this equation, which contains no values referring to the working medium, we can eliminate the quantities $c\,W$ and $r\,K$ by the former equations and then have

$$Q_o\,\frac{\log nat\,T_o - \log nat\,T_l}{T_o - T_l} = \frac{Q_2}{T_l} \quad \ldots\ldots\ldots \quad (288)$$

Absolute conversion-factor.

16) If the process is taking place under permanent conditions, the greatest amount of energy L_1 theoretically obtainable (see Eq. in § 7, 5, p. 19) is

$$A\,L_1 = Q_o - Q_2$$

and by combining this equation with Eq. 288, we get

$$A L_1 = Q_0 \left(1 - T_l \frac{\log nat\, T_0 - \log nat\, T_l}{T_0 - T_l}\right) \ldots \ldots (289)$$

as the required factor of conversion expressing the fraction of the heat-quantity Q_0 which under the existing circumstances can be converted into work with a fall of temperature from T_0 to T_l.

17) In boilers however, the production of work does not begin at the combustion temperature of the furnace gases, but in the most favourable case only at the temperature of the boiler steam and the range between these two temperatures is lost for the transformation of energy, which circumstance is a defect in all steam boilers. Neither is the whole heat Q_0 imparted to the steam, because the hot gases do not enter the chimney at temperature T_l but at T_2 and are besides exposed to radiation losses. It has been invariably the custom hitherto to represent the heat Q_1 actually supplied to the steam as a fraction of Q_0 which should be "theoretically" communicated to it, leaving quite out of account the imperfect application of the range of temperature. This fraction which takes all radiation losses into account will be here called the coefficient of heat-transmission φ; **Actual process.**

then $$Q_1 = \varphi\, Q_0.$$

On account of the radiation-losses φ is always $< \dfrac{T_0 - T_2}{T_0 - T_l}$.

For the further course of the cycle only the heat Q_1 is to be considered and treated as if it were all taken up by the working medium at the constant temperature T_s. During the transmission of the heat to the cooling body (the air) nothing is changed but the quantity of heat itself. In practice the lower temperature-limit is considerably raised, a fault due to the condenser alone and not to the boiler, so that it does not require to be regarded here. The most advantageous cycle under these circumstances is CARNOT'S for which, if Q_1' is the heat to be rejected at the lower temperature T_l, this simple relation exists

$$\frac{Q_1}{T_s} = \frac{Q_1'}{T_l}.$$

18) The work L_2, here obtainable in the most favourable case, follows however from the equation for the permanent condition **Actual factor of conversion.**

$$A L_2 = Q_1 - Q_1'$$

or, referring to the last two equations

$$A L_2 = Q_1 \frac{T_s - T_l}{T_s} = \varphi\, Q_0 \frac{T_s - T_l}{T_s} \ldots \ldots (290)$$

This expresses the work still obtainable, after the introduction

of the boiler into the process, as compared with the gross factor of conversion in Eq. 289.

Thermodynamic Efficiency. 19) The quotient of these two values represents the required thermodynamic efficiency which affords the only criterion of the use made of the energy supplied to the boiler. With reference to the value for Q_1, this efficiency is obtained from Eqq. (289) and (290) as

$$\eta_t = \frac{L_2}{L_1} = \varphi \frac{T_s - T_l}{T_s\left(1 - T_l \dfrac{\log nat\, T_o - \log nat\, T_l}{T_o - T_l}\right)} = \varphi \frac{(T_s - T_l)(T_o - T_l)}{T_s\left(T_o - T_l + T_l \log nat \dfrac{T_o}{T_l}\right)} \quad (291)$$

Example. 20) *Example*, let the temperature above the grate be

$t_o = 1400^0$, therefore $T_o = 1673^0$,

the temperature of the uptake gases

$t_2 = 300^0$, therefore $T_2 = 573^0$,

the temperature of the external air

$t_l = 15^0$, therefore $T_l = 288^0$,

the temperature of the dry steam supplied to the engine $t_s = 200^0$, therefore $T_s = 473^0$,

and let the radiation-loss be $10\,^0/_0$ of the heat imparted to the boiler (see table on p. 206).

In the first place the coefficient of conversion is by 17)

$$\varphi = 0.9\,\frac{1673 - 573}{1673 - 288} = 0.715$$

and the absolute factor of conversion of heat is by Eq. 289

$$A L_1 = Q_o\left(1 - 288\,\frac{\log nat\ 1673 - \log nat\ 288}{1673 - 288}\right) = 0.635\, Q_o.$$

But by the preceding this range of temperature and consequently this energy is not available for use in the engine in consequence of the insertion of the boiler into the system, as even when the lower limit of temperature is a horizontal straight line the factor of conversion

$$A L_2 = \varphi\, Q_o\,\frac{473 - 288}{473} = 0.391\ \varphi\, Q_o = 0.280\, Q_o,$$

so that the thermodynamic efficiency of the boiler in question becomes

$$\eta_t = \frac{L_2}{L_1} = \frac{0.280}{0.635} = 0.442.$$

Value of the investigation. 21) From the investigation of the thermodynamic efficiency of boilers it is evident that the effect of the boiler upon the conversion of energy is that not only a portion of the available heat is lost, but that the factor of conversion (of the remainder) i. e. that fraction of it which is at the best capable of being transformed into work, is considerably di-

minished. It would therefore be a mistake to regard the engine as alone responsible for the unfavourable use made of the heat liberated by the combustion, because it is supplied to the engine with a much lower factor of conversion than to the boiler. For the thermodynamic efficiency of marine engines see § 17, 39.

§ 54.
Box Boilers.

1) **I. Classification and Design.** Marine boilers of the present day may be divided into the following groups, viz. Classification according to pressure.

Box boilers up to 4 atmos. working pressure,

Cylindrical and locomotive boilers up to 15 atmos. working pressure,

Water-tube boilers of above 15 atmos. working pressure.

2) **Box boilers** have large flat sides and their fundamental figure is that of a four-sided prism, sometimes bevelled or rounded off at the top or bottom to suit the form of the ship. All box boilers still at work have tubes above their furnaces. Box boilers with their tubes placed at the sides of the furnaces as they were formerly fitted in small war-ships of light draught have become gradually disused and are no longer constructed. Classification of Box boilers.

3) Box boilers with tubes above the furnaces are of two kinds, Kinds of box boilers.

a) flat-bottomed, and

b) dry-bottomed.

4) a. **Flat-bottomed boilers** were designed in the early days for wooden steamers. To avoid danger to the hull the furnaces had to be kept entirely within the shell. Figs. 1 and 2, Pl. 19 are illustrations of such boilers. Flat-bottomed boilers.

5) b. **Dry-bottomed boilers** were only used on board iron ships. They were mostly fitted in armoured vessels, the double bottom entailing such shallow bilges that it was difficult to keep flat-bottomed boilers properly protected from rust underneath, and to avoid this danger of rust the bottom surface was reduced as much as possible which resulted in the boiler shewn in Figs. 3 and 4, Pl. 19. Dry-bottomed boilers.

6) The boilers of both types were alike in their construction and arrangement, the number of furnaces varied between 2 and 5. Of course those of higher working pressure had rather heavier plating and stays than the others. So long as the working pressure remained low both kinds gave satisfactory results which were attributed to the large roomy furnaces and com- General arrangement of Box boilers.

bustion chambers that could be got in. The furnace crowns could be placed so high above the grate that the combustion gases had plenty of room to mix properly with air while the ash-pits could be placed sufficiently low for ready access of air beneath the bars. From the furnaces the gases passed to the combustion chamber and thence through the tubes to the smoke-box and the funnel. The tubes were usually somewhat inclined upwards from the back to the front, to assist the draught. The working pressure was between 2 and 4 atmos. and the engines were either single-expansion or medium-pressure compounds. These boilers as a rule had superheaters.

Advantages and disadvantages of box boilers.

7) At the same working pressure box boilers require less room in the ship for equal power than other boilers and have also more steam space in proportion to their grate and heating surface. On account of their large water space forming a reservoir of heat they can, within limits, bear irregular firing and feeding and still keep a regular pressure. They were therefore very easy to work. The disadvantage of them was that as the working pressure became higher, their flat sides required so much staying as to render them too heavy and expensive.

Economy of box boilers.

8) **II. Abandonment of Box boilers.** When the first high-pressure engines described in § 44, 9 were introduced for steamers, box boilers were still in use and they were even originally retained for compounds up to 4 atmos. working pressure. It was soon found out that low-pressure box boilers, with their roomy furnaces, so favourable to good combustion, were more economical steam-generators than the cylindrical boilers of that day with furnaces only a little over 0.8 m diameter and often very long besides. The average cross-section of the combustion space over the grate in the old box boilers was almost 33% greater than in the newly introduced cylindrical boilers. And here it must be specially noted that the usual length of fire-bars in the box boilers was only 1.75 m against 2 m in the cylindrical ones. These altogether more favourable circumstances of the furnaces in the box boilers made them 5 to 6% more economical than the cylindrical boilers, — a matter of general experience (compare § 44, 12).

Strengthening of the surfaces.

9) From all this it follows that the principal marine engineers of that day were not to be blamed for trying to adhere to the more economical type of boiler to which their boiler-makers were thoroughly accustomed by years of practice, a most important point for the manufacturer. Besides, the locomotive fire-box had demonstrated that flat surfaces in boilers were perfectly well able to withstand high pressures if sufficiently

stayed. They are even employed today in boilers of this type up to 15 atmos. working pressure, or about 4 times that of the old box boilers which were however submitted to a water-test of 8 atmos. pressure. The oval boilers introduced later became very popular and never gave any trouble with their flat sides. In the box boilers the plates exposed to the fire had to be increased in thickness on account of the rise in the pressure. Whereas 9.5 mm had been their usual thickness in the mercantile marine, it was afterwards made 13 to 14 mm or equal to that of the modern corrugated furnace.

10) The marine engineers of that day*) were censured for thus increasing the thickness of plates in contact with the fire because it was assumed, without the slightest evidence, that such thick plates would greatly detract from the transmission of heat. It has since been proved by means of certain marine boilers designed by KILVINGTON and TAYLOR of Wallsend-on-Tyne which have been at work for a great number of years, that cylindrical furnaces up to 19 mm in thickness shew no diminution in conductivity. Upon the author's quoting this fact from the Transactions of the North East Coast Institution of Engineers and Shipbuilders**), he was furnished by Mr. BACH with the Report of the Württemberg Boiler Inspection Association for the year 1891, from which it appears that they have excellent stationary boilers under their care with furnaces up to 1750 mm in diameter and 23 mm in thickness, the conductivity of which is entirely satisfactory. BACH very appositely remarks that *"the thickness of the furnace plates has nothing like to much influence upon their conductivity as it was hitherto assumed to have"*. **Slight diminution of conductivity with increasing thickness.**

11) **The reasons why high pressure box boilers were given up** had more to do with practice than design; they were chiefly **Reasons for the abandonment of box boilers.**

a) bad workmanship and
b) bad treatment at sea.

12) **The bad workmanship** consisted in employing the same system of work on the high pressure box boilers as formerly on the low pressure ones. If the present methods of boiler-making had been adopted then, i. e. if all the holes had been drilled in place and countersunk inside and out, all the corners carefully fitted hot, &c, the original leaks would have been prevented. **Bad Workmanship.**

*) Compare SCHWARZ-FLEMMING'S remarks in "Die Kesselabtheilung auf Dampfschiffen". 1873, pp. 16 and 40 which exactly represent the entirely erroneous views of a great portion of the sea-going engineers of the day.

**) Zeitschrift des Vereins deutscher Ingenieure. 1891, p. 93.

Bad treatment.

13) **The bad treatment** arose from the fact that nobody thought of working the boilers any the more carefully on account of their higher pressure and temperature. It was not only customary to get up steam in the very short time of $1^1/_2$ to 2 hours, but to let the boilers cool down suddenly at the end of every run, as immediately after the fires were drawn the water was generally blown out either entirely or in more favourable cases down to the furnace crowns. The care which now-a-days the most negligent engineer devotes to his boilers was then a rare exception. It is therefore only natural that these badly constructed and originally leaky boilers should always be getting leakier still in consequence of the strains set up by rapid heating and cooling.

Disadvantages in working.

14) The loss of water by leakage necessitated a large amount of supplementary feed (which of course brought the temperature of the feed-water pretty low) as well as very frequent blowing off in consequence of the rapidly increasing saltness of the water with a boiler temperature of over 144° C. Another great objection to these boilers was the difficulty of getting about inside them owing to the great number and close pitch of the stays. The heavy deposits of salt and scale which were inevitable with such a system of working the boilers necessitated their being thoroughly cleaned at frequent intervals. This however could not be done after every voyage because of the attendant delay. Thick accumulations formed on the tubes, furnaces, and combustion chambers and besides greatly increasing the consumption, were constantly causing serious interruptions in the working. After these experiences box boilers for high pressure and compound engines had to be given up, although they were so economical in principle and so convenient to fit in the ship.

Actual efficiency.

15) **Efficiency of Box boilers.** a. **Actual Efficiency.** The best of the single expansion and medium-pressure compound engines required about 13 kg of water per horse per hour with box boilers. As the most efficient of these boilers produced an average of 100 *IHP* per sq. m of grate at forced trials, burning about 150 kg of coal per sq. m of grate per hour, the weight of water evaporated per kg of coal comes out

$$\frac{100 \times 13}{150} = 8.7 \text{ kg}.$$

Assuming 1 kg of good coal to be theoretically capable of converting 14.5 kg of water into steam of low pressure (compare § 20, 22) the actual efficiency is

$$\frac{8.7}{14.5} = 0.60.$$

16) b. **Efficiency according to Rankine.** Box boilers have about 30 sq. m of heating surface per sq. m of grate and therefore a calculated efficiency of Calculated Efficiency.

$$\frac{Q_1}{Q_0} = 0.92 \frac{30}{30 + 0.1 \times 150} = 0.61.$$

§ 55.

Cylindrical Boilers.

1) **I. Division.** In most cases marine cylindrical boilers are placed horizontally. Vertical boilers are only used for river steamers and as auxiliary boilers on sea-going ships. General Classification.

2) *Horizontal cylindrical boilers* may be classified according to the position of their tubes and the route by which the furnace gases pass from the grate to the chimney, as Division of horizontal cylindrical Boilers.

a) boilers with through tubes or Navy type boilers, and
b) „ „ return „ „ Scotch boilers.

3) **II. Navy-type boilers** have tubes running parallel to the centre line of the boiler from the combustion chamber to the smoke-box which is at the back end, so that the gases traverse the boiler once from end to end. This design originated from the locomotive boiler described in the next paragraph. Navy type boilers are now almost exclusively used in war-ships of small dimensions and light draught, where they have to be kept as much as possible below the waterline to protect them from shot, or where the confined space beneath an armoured deck will not admit boilers of greater diameter. On the adoption of higher working pressures for war-ships these boilers replaced the old shallow box boilers with tubes at the side of the furnaces. (See Pl. 3 of the first German edition of this work.) As Plates 20 and 21 shew, they have been constructed quite recently with one, two, and three furnaces. To what purposes applied.

4) The first boilers of this description intended for high pressure twin engines and shewn in Figs. 1 and 2, Pl. 20 did not become celebrated for their steaming qualities. The furnace was much too small in diameter and the combustion chamber too short for satisfactory combustion. The system of putting the boilers together with angle-irons shews how completely the methods of construction which had grown up with the box boiler were transferred to the cylindrical one. Boilers with one furnace.

5) Considerable progress was exhibited by the two-furnace boilers of the German dispatch-boat "Zieten" Pl. 20, Figs. 3 and 4, the furnaces having a good diameter and the combustion- Older boilers with 2 furnaces.

chamber the necessary length which experience shews to be about 1 m. Hanging from the combustion-chamber top there is a second bridge against which the hot gases impinge and are then diverted beneath it, thus becoming more thoroughly mixed with the air. They also distribute themselves better over the tubes, whereas if this hanging bridge is not fitted the greater portion of them passes through the upper tubes and the lower ones lose some of their efficiency as heating surface. When the fire-door is opened this hanging bridge assists the fire-bridge in preventing the direct access of the cold air to the hot tube-plate, thus diminishing the risk of leaky tubes. On the other hand, it cannot be denied that the second bridge somewhat interferes with the draught especially when the fires are forced, so that although this bridge undoubtedly increases the evaporative efficiency it also detracts from the rapidity of making steam.

More recent Boilers with 2 furnaces.

6) Further examples of this type of boiler are those of the British gun-boats of the "Thrush" class of 10 atmos. working pressure, Figs. 1 and 2, Pl. 21. These boilers are entirely of steel and have corrugated furnaces instead of the former plain ones. Similar boilers with an ash tube but plain furnaces, were first fitted towards the end of the seventies to the British corvettes of the "Comus" class. In place of the four longitudinal stays fitted below the tubes in the "Zieten" boilers, these boilers had a large tube enabling ashes, clinker, &c to be withdrawn from the combustion chamber under steam. This tube has a lid at its external end, which is only lifted when ashes are to be drawn.

Boilers with 3 furnaces.

7) Since the beginning of the eighties large navy-type boilers have been designed with three furnaces, as exemplified in Figs. 5 and 6*), Pl. 20 and Figs. 3 and 4, Pl. 21. But in these boilers the ash-tube had to be omitted on account of the middle furnace. Probably one of the chief reasons for constructing these large boilers of the Navy type instead of with return tubes was their extra safety in bad weather, there being less danger of the combustion-chamber crowns getting hot for want of water when the ship heels over than in return-tube boilers. These boilers with three furnaces, as well as those of the same type with two, gave rise to the question whether it would not be better for the combustion to fit a separate combustion-chamber

*) It did not come to the Author's knowledge till after the plates were printed that the American Bureau of Construction had substituted Scotch boilers for those originally designed for the "Katahdin" here illustrated.

to each furnace, instead of one, common to all the furnaces. On this question however, the Commission referred to below*) were unable to agree, the majority deciding against separate combustion chambers, probably only for the sake of saving weight. How very great importance is now attached to this point in naval boilers and how it has brought about a reduction in the thickness of shell plating of steel boilers in spite of increased pressures and diameters, is shewn in the following short table of particulars of several boilers of this type in the British Navy.

Ship's name	Year built	Working Pressure in atmos	External diam. of Boilers in m	Thickness of Shell plating
"Warspite"	1881	6.33	2.85	19
"Surprise"	1884	7.00	3.15	16
"Thrush"	1889	10.00	2.38	15

8) In other respects the Commission arrived at the conclusion that the boilers of the navy gave very good results and that the usual ratio of grate-surface to heating surface (generally 1 : 32) as well as that of steam-space to water-space fulfil the requirements of practice. It was however specially pointed out that the water-spaces round combustion-chambers should not be made narrower than in the "Thrush" class, otherwise the circulation becomes retarded. Finding of the Commission.

9) **III. Efficiency of Navy Boilers.** Below will be found both for the older and more recent boilers of this type Efficiency.

a) the actual efficiency,
b) the efficiency according to RANKINE,
c) „ „ „ „ WILSON.

10) a. **The actual efficiency** is calculated for the older navy boilers (Pl. 20) as follows. Usually the working pressure is from 4 to 7 atmos. and the engines are surface condensing single-expansion or compound. The average water per horse per hour is 10 kg. On forced trials 135 kg of coal have been burnt and 110 *IHP* obtained per sq. m of grate per hour, thus the water evaporated per kg of coal is Actual efficiency of the older Navy Boilers.

$$\frac{110 \times 10}{135} = 8.2 \text{ kg}$$

Assuming 1 kg of good coal, which theoretically converts 14.5 kg of water into low pressure steam (compare § 20, 22), to be only able to evaporate 14.4 kg of water, owing to the

*) Conclusions and recommendations of the committee appointed by the Lords Commissioners of the Admiralty to consider existing types and designs of propelling machinery and boilers in H. M. Ships. London 1892.

higher pressure, we get the actual efficiency of these older navy boilers as

$$\frac{8.2}{14.4} = 0.57.$$

Actual efficiency of recent navy boilers.

11) Recent navy boilers (Pl. 21) mostly have a working pressure of 10 to 12 atmos. and actuate triple-expansion engines. The average water consumed by these engines is 7.5 kg per horse per hour. On account of the greater heating surface of the recent boilers and their consequently somewhat less active draught, they only burn 120 kg per sq. m of grate at forced trials under natural draught, producing 130 *IHP* per sq. m of grate per hour, so that the evaporation per kg of coal is

$$\frac{130 \times 7.5}{120} = 8.1 \text{ kg}.$$

If 1 kg of good coal theoretically evaporates 14 kg of water at this increased pressure, then the actual efficiency of the recent navy boilers is

$$\frac{8.1}{14} = 0.58.$$

Calculated efficiency according to Rankine.

12) b. **The calculated efficiency according to Rankine** for the older navy boilers with a ratio of heating surface to grate-surface of about 25 : 1 comes out

$$\frac{Q_1}{Q_0} = 0.92 \times \frac{25}{25 + 0.1 \times 135} = 0.60.$$

In the later navy boilers the ratio of heating surface to grate-surface is generally taken at 32 : 1, so that

$$\frac{Q_1}{Q_0} = 0.92 \times \frac{32}{32 + 0.1 \times 120} = 0.67.$$

The calculated efficiency according to Wilson.

13) c. **The calculated efficiency according to Wilson**, on reference to § 53, 13, will be found for the older boilers to be

$$\frac{Q_1}{Q_0} = 0.8 \times \frac{25}{25 + 0.1 \times 135} = 0.52$$

and for recent boilers of the same type

$$\frac{Q_1}{Q_0} = 0.8 \times \frac{32}{32 + 0.1 \times 120} = 0.58.$$

Value of Navy Boilers.

14) Navy boilers possess the advantages not only of being easily got into a protected position in small war-ships, but also of making steam more rapidly and being better able to stand forcing than return-tube boilers. The disadvantages which keep them out of merchant steamers are principally their great length and the large space they occupy in proportion to their heating surface. Finally they have the drawback of requiring, besides the ordinary stokehole, a space at the back-end for

cleaning the tubes which entails a sacrifice of much cargo-room in merchant steamers.

15) **IV. Cylindrical Boilers with return-tubes, otherwise called Scotch Boilers** are now the most usually fitted both in war and merchant ships. They are of two kinds, distinguished by the arrangement of the furnaces, viz. Division.

a) single-ended boilers,
b) double „ „ .

16) a. **Single-ended Boilers** are only fired from one end, double-ended boilers from both. In the single-ended Scotch boilers, Plates 22 to 24, the tubes are above the furnaces and the hot gases return from the combustion chambers to the front. Single-ended boilers have from one to four furnaces, according to the diameter of the shell. In small boilers up to about 2.5 m diameter, one furnace is considered sufficient, for diameters between 2.5 and 4 m, two furnaces, and boilers above 4 m have three. Scotch boilers have been made up to 5 m diameter and a little over, four furnaces being put in when the diameter reaches about 4.5 m. It has already been pointed out in § 21, 7 that, having regard to more perfect combustion it is advisable to fit two roomy furnaces rather than three smaller ones, and the same reason makes three preferable to four. The British Commission before referred to therefore recommends four furnaces for single-ended boilers only in cases where there is not sufficient space to get in the necessary number of three-furnace boilers, or where the furnace diameters would come out too large if only three were fitted. Design.

17) Although few and large furnaces are in general preferable to many and small ones, it is nevertheless sometimes a necessity to adopt the latter arrangement when it is impossible by any other means to get the required grate-surface into the boiler and preserve a proper ratio between grate and heating surface. The only alternative would be to make the grates which are already very wide more than from 1.8 to 2 m long, or so large as to be too much for even the most vigorous and skilful firemen. Another objection to few and large furnaces is the great body of dead water there is below them, entailing the constant use of some circulating arrangement to keep the boilers uniformly warmed. Disadvantages of few furnaces of large diameter.

18) A much more burning question than the fittest number of furnaces is whether the combustion chambers should be common or separate. In boilers with two furnaces (Figs. 1 and 2, Pl. 22) the combustion chambers are mostly separate, enabling a tube to be stopped without drawing both fires. They are also separate Common and separate combustion chambers.

in three-furnace boilers (Pl. 24) as better combustion is thus obtained than with a common combustion-chamber. Four-furnace boilers have been fitted with four combustion-chambers as recommended by the British commission as well as with three and two. With three combustion-chambers the centre one is made common to the two centre furnaces as shewn in Fig. 3, Pl. 23 and where two are fitted each one is made common to two furnaces. Common combustion-chambers whether in greater or smaller number, are not always fitted in order to economize weight and cost, but in many cases because if the combustion chambers were all separate the heating surface would come out too high for the grate-surface which could be conveniently got into the boiler.

Plates.

19) It is to be noted that of the Scotch boilers shewn in Plates 22 to 24, those of "Pfeil", "Etna", "Geyser", "St. Rognvald", and "Marie Henriette" drive compounds at 5 to 7 atmos., and those of "Wõrth" and "John Cockerill" triples at 12 atmos. The engines of "Pfeil" and "Blitz" are of the same power and are entered in the table on p. 416 together with those of "Wõrth", those of "Marie Henriette" are in the table on p. 306. "Etna's" boiler (Pl. 22) is, on account of its shortness, an excellent example of a cargo boat's boiler; it takes up little floor-space and therefore encroaches but slightly on the stowage capacity, while the short grates are very handy to fire. "Geyser's"*) boiler (Figs. 1 and 2, Pl. 23) exhibiting the prevailing type at that time fitted in the North American lake steamers, has the objectionable feature of much too long tubes, giving rise to perpetual leaks, because with the area of heating surface to grate of 38 to 1, artificial draught is a necessity. The boiler is only mentioned here in order to shew that it is not good to adopt an exaggerated length. "St. Rognvald's"**) boiler (Figs. 3 and 4, Pl. 23) is of the usual type in England in the middle of the eighties for compound engines. Two such boilers were fitted, with their axes fore and aft, the stokehole running athwartships between them. Each of the boilers produced 750 *IHP*. "Wõrth's" boilers gave 143.4 *IHP* per sq. m of grate during the forced trial with 28 mm air pressure in a closed stokehole. The results obtained from "John Cockerill's" two boilers with Serve tubes are referred to in § 65.

Number of furnaces.

20) b. **The Double-ended Boiler** is the same thing as two single-ended boilers united at their backs and having their back plates

*) W. Cowles. Paper read before the American society of mechanical engineers, New York 1886.

**) A. Jamieson. A text book on steam and steam engines. London 1886. p. 316.

omitted. They have been constructed with from 2 to 5 furnaces in each end; 3 or 4 are however the most usual, as shewn in Plates 25 to 27. The remarks on the number of furnaces in single-ended boilers apply equally to double-ended ones. Three furnaces in each end is therefore the number most to be recommended. The large double-enders with 10 furnaces (Plate 25, Figs. 1 and 2) fitted by PALMERS in 1870 to the Guion liners "Wisconsin" und "Wyoming" are probably a solitary case. Being about 5.2 m in diameter and 7.3 m long, they are among the largest, if not actually the largest, marine boilers hitherto made. Except for the moving fire-bars originally fitted to them and soon replaced with ordinary ones, these boilers answered well. They remained eighteen years in the ships, which must be considered highly satisfactory under the severe conditions of the transatlantic mail service.

21) The ten furnaces of "Wisconsin's" boilers were all connected to one common combustion chamber having four nests of tubes to each end. Six circulating tubes with internal down-flow tubes were fitted in the combustion-chamber to assist the circulation, as experience had shewn that although these large combustion-chambers common to all the furnaces made the boilers lighter and cheaper, they rendered them especially liable to priming unless some circulating arrangement were fitted; they are also less favourable to combustion than combustion-chambers common to two opposite furnaces, as represented in Figs. 3 and 4, Pl. 25, Figs. 1, 2, 5, and 6, Pl. 26, and Figs. 1 and 2, Pl. 27. This last arrangement has proved its fitness for the hard service of transatlantic mail steamers, as "Arizona", "Saale", "Lessing", and "Fürst Bismarck". There are engineers however who are in favour of separating the two opposite combustion-chambers, although the boilers are thus rendered longer and heavier, and uniting either all (Figs. 5 and 6, Pl. 25), or every two (Figs. 3 and 4, Pl. 27) furnaces at each end of the boiler in one common combustion-chamber. The British Commission favours one separate combustion-chamber to each furnace only, unless considerations of weight interfere. But apart from their greater weight, boilers thus fitted have the drawback of requiring to be very gently warmed up and evenly fired, or unequal expansion between the plates of the combustion-chambers and the boiler ends will occur, distorting the plates and causing the tubes and stays to leak. These boilers are also awkward to clean and finally there is a certain amount of difficulty in getting sufficient grate surface into them to be in proportion to their large heating surface. But they have this advantage that while Combustion-chambers.

one fire is being cleaned, the cold air only gets into one combustion-chamber and nest of tubes and therefore does not materially diminish the steaming power of the boiler. For similar reasons, a tube becoming leaky matters much less than with combustion-chambers common to several furnaces.

Advantages and Disadvantages of Single and Double-ended Boilers.

22) On comparing the merits and demerits of single and double-ended boilers, it is apparent that single-enders must be preferable for war-ships, if only because they admit of a finer gradation of power from cruising speed to full speed than double-enders, as emphasized in the British Commission's report. They are however heavier and more costly than double-enders and require more space with a given heating surface. Being so much shorter they can adapt themselves more easily to the strains set up by expansion and therefore need less careful handling than double-enders. But for merchant ships double-enders are always the most advantageous because of their smaller cost and volume. When their diameter exceeds 4.75 m and their length 6 m they are very difficult to handle in the shop as well as to place on board, and can only be renewed where powerful cranes or sheer-legs are available.

Efficiency of Scotch Boilers.

23) **V. Efficiency of Scotch Boilers.** With these boilers also there is a difference between the efficiency of the older ones which were more or less considerately worked and the modern ones which are often much forced, as is shewn by the following figures.

Actual Efficiency of the older Scotch Boilers.

24) a. **Actual Efficiency.** The older Scotch boilers (Pl. 25) are rather behind the older Navy boilers in rapidity of steaming. The former can only burn on natural draught trials 105 kg of coal per sq. m of grate per hour against 135 in the latter; the former only give about 85 *IHP* per sq. m of grate per hour and the latter 110. These older Scotch boilers are mostly of 4 to 7 atmos. working pressure and drive surface-condensing compound engines. When superheaters are fitted they are only used as steam-driers. As the engines are estimated to require on an average 10 kg of water per horse per hour, 1 kg of coal evaporates

$$\frac{85 \times 10}{105} = 8.1 \text{ kg of water,}$$

Their actual efficiency is therefore

$$\frac{8.1}{14.4} = 0.56.$$

Actual Efficiency of recent Scotch Boilers.

25) Modern Scotch boilers (Pl. 27) are worked at 10 to 13 atmos. and almost exclusively with triple-expansion engines. On trial-trips the water per horse per hour of the best of these engines is about 6.5 kg. The coal burnt hourly and the *IHP* produced per

sq. m of grate are 110 kg and 140 *IHP* respectively. Thus 1 kg of coal evaporates

$$\frac{140 \times 6.5}{110} = 8.3 \text{ kg of water.}$$

Assuming that 1 kg of coal can evaporate 14 kg of water at these pressures, the actual efficiency of the boilers is

$$\frac{8.3}{14} = 0.60.$$

26) b. **The Calculated Efficiency according to Rankine,** with the same ratio of heating surface to grate as was usual in the earlier Navy boilers, is Calculated Efficiency according to Rankine.

$$\frac{Q_1}{Q_0} = 0.92 \times \frac{25}{25 + 0.1 \times 105} = 0.65.$$

In recent Scotch boilers the ratio of heating surface to grate is mostly higher than in the old ones, viz. 33 to 1, so that RANKINE'S efficiency is

$$\frac{Q_1}{Q_0} = 0.92 \times \frac{33}{33 + 0.1 \times 110} = 0.69.$$

27) c. **The calculated efficiency according to Wilson** for the older Scotch boilers, if the coefficient B is taken at 0.8, is Efficiency according to Wilson.

$$\frac{Q_1}{Q_0} = 0.8 \times \frac{25}{25 + 0.1 \times 105} = 0.56.$$

and for the recent ones

$$\frac{Q_1}{Q_0} = 0.8 \times \frac{33}{33 + 0.1 \times 110} = 0.60.$$

28) **VI. The Oval Boiler** may be described as a cylindrical boiler cut in halves by a horizontal plane passing through its axis and having then a prismatic piece up to 1 m deep inserted between the two halves. In other respects it exactly resembles a horizontal cylindrical return-tube boiler. These boilers have also been built double-ended both for naval and mercantile purposes (see Figs. 5 and 6, Pl. 25 *) and were specially adapted for narrow ships where floor-area was more important than height. They are mostly fired fore and aft, as in "Arizona", the example here selected. They generaly have 2 furnaces in each end, although the above vessel's boilers have 3. The same considerations as to the arrangement of their combustion-chambers apply as in the case of single and double-ended cylindrical boilers. As their flat sides require a number of stays, the top and bottom rows of which coincide with the boundaries of the flat surfaces, these boilers come out rather heavier than cylindrical ones. But they have the advantage of a higher and larger steam-space, Oval Boilers.

*) Engineering 1880. II. p. 496.

so that they can supply drier steam. Oval boilers are now obsolete, they could only be used under working pressures suitable to compounds as it is hardly practicable to efficiently stay their flat sides for the present high pressures.

§ 56.

Locomotive Boilers.

Object.

1) **I. Object and Classification.** Torpedo-boats, catchers, fast launches, and similar vessels require very light and efficient boilers for their high speed and low displacement. The marine engineer, on looking round for a suitable boiler, naturally lighted upon that of the locomotive type as a model proved by experience to combine high power with compactness. Although this boiler was at first adopted almost unaltered, it has in the course of time undergone certain changes fitting it more closely for marine requirements.

Classification.

2) The marine boilers of locomotive type now in use may be distinguished as

dry-bottomed and

wet-bottomed.

Constituent Parts.

3) **II. The Dry-bottomed Locomotive Boiler** has been the most used in the various navies. It consists chiefly of

a) a cylindrical barrel enclosing the tubes, and

b) a firebox, open at the bottom and of more or less prismatic form, enclosed in a similarly shaped shell connected to the barrel.

Barrel.

4) a. **The Barrel** has usually a diameter of only 1.5 to 2 m which allows of comparatively thin shell-plating even with high working pressures, thus saving weight. As shewn in Plates 28 and 29, the barrel of the early locomotive boilers was considerably longer than the fire-box. By degrees however the barrels were designed shorter and the fire-boxes longer in order to get in more grate surface. In the largest and newest boilers with nearly 6 sq. m of grate surface the length of the fire-box even exceeds that of the barrel. Although the power of the boilers has thus been augmented their economy has fallen in a similar degree, because the heating surface has not been increased in proportion to the grate and the hot gases leave the boiler at a very high temperature, especially on forced trials. On account of its shortness, the barrel has lately often been made with one course of plating (Pl. 29, Figs 2 and 4) but it usually has two (Pl. 28, Fig. 2 and 7 and Pl. 29, Fig. 6) and rarely three (Pl. 28, Fig. 4). The barrel contains in its lower

part the tubes and in its upper part the longitudinal stays, unless the excellent system of staying represented in Pl. 28, Figs. 3 and 4 and Pl. 29, Figs. 5 and 6, is adopted, combining the necessary strength with internal accessibility of the boiler.

5) b. **The fire-box shell** surrounds the fire-box, its two flat sides being either parallel or somewhat diverging downwards in order to increase the width of the grate (Pl. 28, Figs. 1 and 6). The crown of the fire-box shell is either half round, forming a continuation of the top of the barrel (Pl. 28, Fig. 3, Pl. 29, Fig. 5) or it is flat with more or less rounded edges (Pl. 28, Figs. 1 and 6). Inside the fire-box shell there is either **Fire-box shell.**

α) a copper or

β) a steel fire-box.

6) *α. Copper Fire-boxes* afford more efficient heating surfaces than steel ones on account of their greater conductivity and are especially useful in boilers which are rather small for their work. These fire-boxes are either quite plain (Pl. 28, Figs. 1 and 2), or they have a separate combustion-chamber behind the bridge (Figs. 3 and 4), or again they are provided with a corrugation in the crown close to the tube-plate. YARROW*) bends the crown plate upwards (Fig. 7), NORMAND**) downwards, so that the deep pocket thus formed acts as a second (hanging) bridge. The object of this corrugation is to afford the tube plate a possibility of expanding upwards when highly heated and afterwards of contracting when cooled down, thus preventing leaky tube-ends as far as possible. The stays in row a in Fig. 7, Pl. 28 are therefore arranged as shewn in detail in Fig. 5 so that they can move upwards with the crown plate. **Copper Fire-boxes.**

7) One objection to copper fire-boxes is their tendency to crack between the bottom row of stays and the top row of tubes in consequence of the repeated effects of expansion and contraction. A further drawback is the ductility of the material which renders it necessary when expanding a tube to insert drifts into the neighbouring tubes to keep the tube plate from giving way between them. But this precaution is mostly dispensed with in practice for want of time, and even if observed cannot prevent the breaking down of the structure of the material by the rollers of the expander. — Locomotives, being often fitted with copper fire-boxes, are always quoted as a model for their design and treatment. But the circumstances of locomotives are more favourable to copper as a material for fire-boxes; **Disadvantages of copper Fire-boxes.**

*) Engineering 1886. II. p. 179.

**) The Engineer 1891. I. p. 83.

theirs are in the first place smaller than marine ones, the tubes are kept lower down, and finally the violent oscillations on the road help the bubbles of steam to disengage themselves from the walls of the fire-box, so that it does not get so hot as in the marine locomotive boiler. Besides all this there is the fact that at the general overhaul of a locomotive which takes place every three or four years, the fire-box is generally renewed which is not so easily done on board ship as it involves taking out the boiler.

Steel Fire-boxes.

8) *β. Steel Fire-boxes*, as shewn in Pl. 29, are coming more and more into use for marine locomotive boilers as the quality of mild steel continues steadily to improve. They are free from the drawbacks detailed above, although now and then they have shewn a tendency to crack. One advantage is that their crowns do not require such closely pitched staying, for experience shews that there is more deposit on the fire-box crown than anywhere else, in high-pressure boilers as in low pressure ones, and widely pitched stays facilitate cleaning. Steel fire-box plates do not become so soon over-heated in consequence of the accumulation of scale or grease and are therefore less liable to bulging between the stays than copper ones. A case of this sort in an Austrian torpedo-boat's boiler (illustrated in Figs. 6 and 7, Pl. 28) was investigated with great thoroughness by BURSTYN.*)

Application.

9) **III. Wet-bottomed Locomotive Boilers** (Figs. 1 to 4, Pl. 29) are of specially English origin and were first constructed by Mr. F. C. MARSHALL**) with only a partial wet bottom. Mr. MARSHALL'S firm, Messrs. Hawthorn Leslie & Co. of Newcastle, afterwards went over to boilers with completely enclosed ash-pits, Figs. 3 and 4, in which the circulation is said to be better than in those with open ash-pits. They found it worth while to secure the advantages possessed by these boilers at the cost of a not inconsiderable increase of weight. Since the middle of the eighties they have fitted them to a number of British, Italian, Spanish, and other torpedo-cruisers and their example has been followed by LAIRD of Birkenhead for Chilian and Argentine torpedo-boats and by the Naval Construction and Armaments Co. of Barrow for British torpedo-gunboats. These wet-bottomed locomotive boilers may have

a) one furnace or
b) two furnaces.

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*) Mittheilungen a. d. Gebiete des Seewesens. Pola 1887. p. 621.

**) Transactions of the Institution of Naval Architects. London 1888, Plates XIV and XV.

10) a. **Locomotive Boilers with one Furnace** were originally the only ones used and answered very well so long as the grate-surface did not exceed 3 sq. m. When this limit was passed the first step was to fit two fire-doors instead of one, to enable the fire to be worked better. Then as in the Navy-type boiler an ash-tube was added for the removal of ashes from behind the bridge (Figs. 3 and 4, Pl. 21) and at last the fire was partially divided by several water-spaces (Figs. 1 and 2, Pl. 29) which were said to improve the circulation incidentally. It became however more and more apparent that the grate area to be worked through a single fire-door should not exceed 2 sq. m and as with the ever increasing power of the boilers, grate-surfaces of 4 sq. m and above were arrived at, it became necessary to subdivide the fires. **Drawbacks.**

11) b. **Locomotive Boilers with two Furnaces** are now constructed not only with wet but also with dry bottoms whenever the grate-surface is appreciably above 4 sq. m. Besides the impossibility of working such large single grates properly, there was the further objection that the operation of cleaning a fire lasted proportionately long and the cold air thus admitted into the boiler not only injuriously affected the tube-plate and tube ends, but caused a considerable fall in the steam. Recent large torpedo-cruisers are therefore nearly all fitted with locomotive boilers having two furnaces. The separation was carried out as shewn in Fig. 1 Pl. 29, in which the two centre fire-box side plates must be conceived to be carried back to the tube-plate. The introduction of this water-leg certainly does away with several vertical rows of tubes and thus a portion of the tube heating surface is lost, but it is more than compensated for by the two additional fire-box sides, COUCHE'S experiments having proved that about six times as much water is evaporated per sq. m per hour on the fire-box surface as on the tube surface. **How carried out.**

12) With regard to the recently built wet-bottomed locomotive boilers of British torpedo-gunboats of the "Jason" class, the British Boiler Commission reported them suitable for the requirements of these vessels. They were however of opinion that for natural draught such boilers must have 0,23 sq. m of heating surface per *IHP*, although the usual amount of steam-space and proportion of heating surface to grate are sufficient. On the other hand the Commission recommends rather wider water-spaces and greater pitch of tubes than are generally adopted, in order that with forced firing the bubbles of steam may rise unobstructed to the surface. Water spaces 150 mm wide at the narrowest part and about 185 mm at the widest are regarded **Present Practice.**

as large enough, while the distance from centre to centre of two tubes in the same vertical row should not be less than 75 mm and the horizontal pitch of two vertical rows with zig-zag spacing should be 136 mm at least.

Efficiency.

13) **IV. Efficiency of Locomotive Boilers.** As in the case of cylindrical boilers, the older locomotive boilers with a single copper fire-box often drove compounds or triples of low working pressures, while modern boilers of this type with two steel fire-boxes work triples or quadruples exclusively. Their efficiency is consequently different. The efficiency of locomotive boilers as determined in the succeding sections refers to conditions of the hardest forced draught and not to natural draught as for cylindrical and box boilers, because locomotive boilers have only been used in warships and are therefore only valued for their steaming capacity, not for their economy.

Actual Efficiency of the older locomotive Boilers.

14) a. **The actual Efficiency** of the earlier locomotive boilers can be determined when we know that their working pressure was from 10 to 12 atmos. and that under forced draught they burnt per hour about 300 kg of coal and developed about 220 *IHP*, per sq. m of grate. The steam consumption at the high cut-offs usual on these forced runs was 8 kg per horse per hour. Therefore 1 kg of coal evaporated

$$\frac{220 \times 8}{300} = 5.86 \text{ kg of water.}$$

Assuming that 1 kg of coal theoretically converts 14 kg of water into steam of these high pressures, we get the actual efficiency

$$\frac{5.86}{14} = 0.42.$$

Actual Efficiency of recent locomotive Boilers.

15) The working pressure of modern locomotive boilers is usually between 13 and 15 atmos. and the triples or quadruples they drive develop about 350 *IHP* and burn per hour about 500 kg of coal per sq. m of grate with heavy forced draught. But the steam consumption is no lower than formerly and must therefore be still taken at 8 kg per *IHP* per hour. Accordingly 1 kg of coal evaporates

$$\frac{350 \times 8}{500} = 5.6 \text{ kg of water}$$

again assuming 14 kg to be the weight of water which 1 kg of coal ought to evaporate at these high pressures, we get for the actual efficiency of recent locomotive boilers

$$\frac{5.6}{14} = 0.40.$$

16) b. **The Efficiency according to Rankine** for the older locomotive boilers with a ratio of heating surface to grate usually about 40 : 1 comes out Efficiency according to Rankine.

$$\frac{Q_1}{Q_0} = 0.92 \frac{40}{40 + 0.1 \times 300} = 0.52.$$

In modern locomotive boilers the ratio of heating surface to grate rises to 60 : 1, so that

$$\frac{Q_1}{Q_0} = 0.92 \frac{60}{60 + 0.1 \times 500} = 0.50.$$

17) b. **The Efficiency according to Wilson** for the older locomotive boilers, — the coefficient being 0.8 instead of 0,92 is Efficiency according to Wilson.

$$\frac{Q_1}{Q_0} = 0{,}8 \frac{40}{40 + 0.1 \times 300} = 0.46,$$

and for recent ones

$$\frac{Q_1}{Q_0} = 0.8 \frac{60}{60 + 0.1 \times 500} = 0.43.$$

In spite of its great heating surface the efficiency of the locomotive boiler is thus only a very low one because of the excessive forcing of the fires. If however these boilers are worked under natural draught with ordinary stoking they are quite as efficient as those of the navy type.

§ 57.

Water-tube Boilers.

1) **Object and Classification.** The desire to produce a powerful, efficient, and compact boiler which, without additional weight or hindrance to good convection, would bear a considerably increased working pressure has led to experiments with water-tube marine boilers which extend back for a long period. While cylindrical boilers require thicker plates and stronger staying as the pressure is increased, water-tube boilers allow of a rise within comparatively wide limits in spite of their light scantling and invariable absence of stays. Against this advantage must be placed all the draw-backs due to the small water-space and inferior circulation of some of these boilers. Of all the water-tube boilers hitherto introduced in Germany, England, France, and the United States, those mentioned below most deserve attention. Object.

2) Marine water-tube boilers may be divided in the first place into two great groups, viz. Straight and curved tubes.

boilers with straight tubes and

" " curved tubes,

their qualities being very different. Each of these groups contains several classes of boilers of similar pattern.

Classification. 3) Passing over launch boilers which will be treated of in § 59, marine water-tube boilers may be classified as follows.

Straight-tube boilers.

A. *Boilers without water-chambers or element boilers.*

a) the PERKINS boiler of 1860,
b) the BELLEVILLE boiler of 1866 and 1878,
c) the HERRESHOFF boiler of 1888.

B. *Boilers with water-chambers.*

d) the ALBAN boiler of 1843,
e) the HEINE boiler of 1877,
f) the ORIOLLE boiler of 1885,
g) the D'ALLEST boiler of 1890,
h) the YARROW boiler of 1892,
i) the MOSHER boiler of 1895,
k) the BABCOCK-WILCOX boiler of 1893.

C. *Water-tube boilers with internal suspended tubes.*

l) the DÜRR boiler of 1883,
m) the NICLAUSSE boiler of 1892.

Water-tube boilers with curved tubes.

D. *Water-tube boilers with upper and lower collectors or drums.*

n) the DU TEMPLE boiler of 1886,
o) the THORNYCROFT boiler of 1887 and 1889,
p) the NORMAND boiler of 1890,
q) the YARROW boiler of 1890,
r) the WHITE boiler of 1883,
s) the THORNYCROFT boiler of 1893.

E. *Water-tube boilers with vertical collectors.*

t) the WARD boiler of 1888.

Older types. 4) In the above classification some early types, like the PALMER, ROOT, WATT, and ROWAN boilers are omitted as having become obsolete in marine practice. They are fully described by the author in a former work.*) Recent boilers, of which only a few have been built, or differing but slightly from those in the list, have also been left out of it, but will be briefly referred to in their appropriate places.

General. 5) **II. Straight-tube Boilers.** The first group of this class, which are without water-chambers, consists of a number of rows of tubes connected together, called *elements*, rising out of a feed-

*) C. BUSLEY. Die Entwickelung der Schiffsmaschine. Edit. III. Berlin 1892. p. 159.

water-container which is common to all of them, and after a zig-zag course uniting in a more or less roomy steam-collector. The principal difference between these boilers lies in the connection between the several rows of tubes, their general structure is pretty much the same.

6) **a. The Perkins boiler of 1860.** The boiler shewn in Figs. 6 to 10, Pl. 30, patented by PERKINS in 1860, was constructed in 1879 and tried in 1880 on board the steam-yacht "Anthracite". It contains a number of continuous wrought iron tubes surrounded by a double casing of thin sheet iron, the 100 mm space between the two skins being filled with wood charcoal, a very bad conductor of heat. **General Arrangement of the boiler.**

7) The boiler has only one furnace surrounded by seven horizontal tubes placed 45 mm apart vertically in the clear. Each of these tubes runs round the furnace and is closed at both ends which are kept 13 mm apart. The fire-door is formed by interrupting two tubes. These seven tubes are connected together by a series of small vertical branches of 33 mm external and 27 mm internal diameter spaced 204 mm apart. The branches are 75 mm long over all and are screwed 15 mm at each end into the horizontal tubes. Above the 7 tubes surrounding the furnace are 140 tubes in 14 vertical rows 45 mm apart and 10 horizontal rows 20 mm apart. Each tube is 1397 mm long also closed at both ends. The ten tubes in each vertical row are connected, close to each of their ends by vertical branches screwed right and left. As the horizontal rows are not connected, the ten tubes in each vertical row form one element by themselves. The bottom tube in each element is connected to the top tube of those surrounding the furnace by two branches cut in the middle and jointed by a screwed union and copper washer. These connections are shewn in Figs. 9 and 10. At the top a pipe of 152 mm external, 101 mm internal diameter, and 1314 mm long is placed across all the tubes 280 mm above them and serves as a steam-collector. It is connected to the top tube of each element by a similar branch to those used at the bottom, so that any element can be taken out without disturbing the rest of the boiler. The steam-pipe of 28.5 mm diameter leads from the steam-collector to the engine and the funnel is placed immediately above the collector. Two cleaning doors are cut in the upper part of the casing and an ash-pit door at the bottom. There is no prescribed water-level, but in the trial referred to below it was kept so that the six upper rows of tubes formed part of the steam-space. **Description of the Boiler.**

Experiment with the boiler.

8) A trial, very elaborately reported on by ISHERWOOD*) was made by LORING in 1880 at the New York Navy Yard whither the "Anthracite" had steamed from England. During the trial which lasted nearly 24 hours the working pressure averaged 22.25 atmos. though its limit was no less than 35 atmos. With approximately 26 expansions the triple-expansion engine developed a mean of 67.7 *IHP* at 1.23 kg of coal per horse per hour. This very mediocre result is explained in the first place by the throttling undergone by the steam, the sectional area of the main steam-pipe being only $\frac{1}{47}$ of the area of the H.P. piston. The wire-drawing was so great that the fall of pressure from the boiler to the H.P. cylinder was 9.14 atmos. at 23.29 atmos. working pressure. The boiler was fed with distilled water made by an evaporator on board from *fresh water*, because the patentee did not consider sea-water pure enough. On the voyage of course sea-water had to be used for the purpose.

Results of the trial.

9) During the trial the hourly average coal burnt per sq. m of grate was 58 to 60 kg and the *IHP* developed was about 48 per sq. m of grate. At this rate while the evaporation per kg of coal was only 9.27 kg of water from and at 100° C, the boiler had already exceeded its limit of efficiency, that is, the combustion was so rapid that the evaporation could no longer keep pace with it. If the combustion had been heightened till 75, 100, or even 120 kg were burnt per sq. m of grate per hour, as in natural draught trials of Scotch boilers, the priming would have emptied the bottom tubes and caused them to get red hot.

Performance of the boiler at sea.

10) On the run across from England to New York the boiler's performance was so low that it only developed 21.5 *IHP* per sq. m of grate per hour or about one-tenth of that obtained in modern marine boilers with forced draught and there was of course no trouble under such extremely easy conditions.

Opinion upon the Perkins Boiler.

11) *The Perkins boiler has not sufficient water line area to allow the steam to pass off and the steam space is too low to cause the water carried up into it to fall back again by its own weight.* If the evaporation becomes comparatively active in consequence of sharp firing, the steam can no longer liberate itself sufficiently rapidly from the water and priming invariably follows, the more aggravated the more the boiler is forced. The PERKINS boiler can therefore only be worked at a comparatively slow rate of evaporation and even then requires a large heating surface, if

*) Journal of the Franklin Institute 1880.

it is to be economical. The performance of this boiler, weighing 7266.5 kg including water, according to ISHERWOOD'S figures, was quite unusually low in spite of the working pressure of 35 atmos. (it was tested to 140 atmos. and each tube separately to 280 atmos. with cold water), for it only developed on the trial 9.4 *IHP* per ton weight of boiler and at sea as little as 4.2 *IHP*, whereas a Scotch boiler can at least double this result with natural draught and a triple expansion engine.

12) **b. The Belleville Boiler of 1866 and 1878.** After BELLEVILLE had in 1864 designed a water-tube boiler with curved tubes, which did not achieve success owing to its confined water-space and the impracticability of cleaning the tubes, he brought out his first straight-tube boilers in 1866. They became known in Germany chiefly through being adopted as the regulation type of launch boiler in the French Navy and were introduced into the German Navy with the steam pinnaces captured in the Loire in 1870/71. **Evolution.**

13) They consisted of several elements composed of horizontal wrought iron tubes placed over each other and connected at their ends by malleable cast-iron sockets. The tubes were connected at the bottom to a rectangular box into which the feed-water passed, and at the top to a similar box from which the steam was taken. The lower half of the tubes was filled with water and the furnace gases passed up from the grate around and among the tubes to the chimney at the top. The boiler casing consisted of two thicknesses of sheet iron, with some bad conductor (common ash) between them. Doors were fitted to the front of the casing, giving access to the tube ends which had loose lids for cleaning purposes. As these steam pinnaces had no surface-condensers they were provided with tanks for fresh feed-water. A drawing of this boiler is given in Figs. 7 to 11, Pl. 6 of the 2nd (German) Edition of this work. **Description of the early boilers.**

14) These boilers were not suitable for sea-going steamers of any size and it was not till 1878 that BELLEVILLE, after many improvements, succeeded in producing the boiler which was first tried on the French dispatch-boat "Voltigeur" and excels all other water-tube boilers in refinement of design. It has consequently been far the most used at sea. Up to the present there are 384 000 horse-power of these boilers built or building for British, French, and Russian war-ships, besides 73 000 horse-power for merchant ships. In the BELLEVILLE boiler of to-day the objections to the older ones are avoided; they were chiefly

α) the very moderate circulation,

β) the small water-space rendering the feeding difficult,

γ) the tendency to prime and produce wet steam,

δ) the difficulty of cleaning.

Improved design.

15) α. *The insufficient circulation* in the old BELLEVILLE boiler with elements consisting of one horizontal row of tubes has been corrected by setting the tubes with a rise of 4^0 and they are now connected in elements of two rows each. Also the head-pieces of the tubes are provided with separating walls and the water striking against these on passing from one tube to the other is mixed and broken up, the hotter water at the outside of the tube uniting with the cooler from the inside, thus diminishing the formation of pockets of steam. This improved boiler is shewn in Figs. 1 and 2, Pl. 30. The connection of each element to the feed-water receiver is effected by means of a cone screwed into the latter and projecting about 15 mm into the head-piece of the element (Fig. 1), the joint being made with a nickel washer about 1 mm thick and 20 to 25 mm diameter slipped on the cone. The element is held down on the cone by a screw passing through a flange on the head-piece. But even in these boilers the circulation is not perfect, for after working a certain time the bottom tubes have always been found to sag more or less. They then have to be unscrewed from the head-pieces and straightened. This sagging occurs although special provision for the expansion of the elements is made by fitting a wrought iron roller 25 to 38 mm diameter working in a small cast iron box beneath the back bottom head piece of each element. According to WATKINSON'S experiments with models*) the circulation in a BELLEVILLE boiler is of a very different character from that in other boilers. The entire expanded length of an element from the feed-water chest to the steam-collector varies in BELLEVILLE'S marine boilers between 30 and 45 m, the top end of each element is about 1.6 to 1.8 m above the bottom one and the connecting pipe to the steam-chest is from 0.3 to 0.5 m long. If all the tubes in one element followed a continuous straight course, a head of about 0.6 m would suffice to drive the water through as in other water-tube boilers. But as a change in the direction of the flow occurs at every head-piece, this pressure is not enough to overcome the resistance. In order to produce satisfactory circulation a non-return valve is fitted on the water-chest and the water is made to pass into the elements through the contracted openings of the cones before described. As the water in the feed-pipe which comes from the steam-chest stands

*) Transactions of the Institution of Naval Architects. London 1896.

at a higher level than in the boiler, it opens the non-return valve, enters the boiler, and is evaporated. But the formation of steam takes place simultaneously in all the tubes, and thus the water is driven upwards by the steam formed in the lower tubes and then forced back on meeting the steam in the upper tubes, a reciprocating flow of the water being caused. The non-return-valve prevents the water from getting out of the boiler and hammering occurs during the process of warming up as in all boilers in which the steam from the tubes enters the steam chest above the water-level. When the BELLEVILLE boiler is fairly at work its circulation is very active and complete, the steam bubbles passing with great rapidity through the tubes.

16) *β. The small water-space,* forming in comparison with that of a Scotch boiler only an insignificant heat-reservoir, does not allow of a sudden change in the demand for steam which is always happening in manœuvring with a fleet. An abrupt increase causes the boilers to prime. To provide against priming BELLEVILLE fits his boilers with a very ingenious, although somewhat sensitive automatic arrangement intended to regulate the feed according to the consumption of steam. It has been found to work well, if properly looked after, even in bad weather at sea, so long as the boiler is not pushed and the demand for steam is regular. This feed-regulator, which works with a special feed-donkey, is described in § 69. Without this pump of BELLEVILLE'S, his boilers have never been able hitherto to be kept at work, which is explained when we learn that the weight of water in the BELLEVILLE boiler is only about one-third of that in the other French water-tube boilers in use on board large steamers. Thus the weight of the boilers complete with all fittings in the mail-steamer "Australien" is 392 tons, that of the water only 20 tons; for the cruiser "Bugeaud" these weights are 273.3 tons and 16 tons, so that the weight of the water is $\frac{1}{17}$ to $\frac{1}{19}$ of that of the boilers. **Special feed-arrangement.**

17) *γ. The wet steam* generated by BELLEVILLE boilers makes them comparatively ill-adapted for responding to an increased call for steam when being *sharply* fired, even under natural draught. It is reported that with very active combustion they are always on the verge of priming and that the gases, comparatively little baffled by the tubes (see Fig. 1, Pl. 30), stream in long bright flames from the funnels. This is in general the same fault that ISHERWOOD found with the PERKINS boiler, which he refers to insufficient water line area. As soon as the boiler is at all pushed the surface of the water is never at rest **Working.**

and if priming does not immediately set in, at any rate wet steam is generated. On this account, as well as because of its inferior combustion, the BELLEVILLE boiler works best under slow evaporation, but as all trial-trips have shown not so well with rapid evaporation. Thus, on the trials of the French dispatch-boat "Milan" with compound engines, only 0.915 kg of coal per horse per hour were burnt at about 60 kg per sq. m of grate per hour, but the consumption rose to 1.08 kg when 130 kg per sq. m were burnt, which is about the highest combustion attained in the BELLEVILLE boiler. At the trials of the French iron-clad "Brennus" with triple-expansion engines 1.79 kg per horse per hour were consumed at first, and later on when the stokers had got into form 1.11 kg, but even the most economical run shewed as high as 0.84 kg. The comparatively bad combustion in BELLEVILLE boilers arises from the fact that a large portion of the gases arrives among the tubes still unburnt. To obviate this trouble compressed air is forced in above the grate by means of special pumps. This compressed air (in small jets) not only affords the gases a better chance of mixing with oxygen, but detains them for a certain time in the fire, thus assisting to burn them more completely. — BELLEVILLE makes use of two devices for drying the steam, a trap placed in the steam-collector to mechanically separate the water from the steam (described in § 73) and a reducing valve in which the already partially dried steam is throttled to about 5 or 7 atmos. below the boiler pressure before it reaches the engine. As steam expanding suddenly without doing work becomes superheated, the intention is by means of this wire-drawing to evaporate the watery particles carried over.

Removal of Deposit.

18) δ. *The difficulty of cleaning* is overcome by BELLEVILLE in the first place by introducing the feed-water through a checkvalve mostly 25 mm diar. into the bottom of the steam-collector and allowing the water to traverse the latter from end to end, the boiler pressure being 20 atmos. and the feed-pump pressure 40 to 45 atmos. As the working pressure in recent BELLEVILLE boilers varies between 12 and 20 atmos., corresponding to a steam-collector temperature of 190° to 215° C., the feed-water entering in a fine jet becomes suddenly very highly heated, the greatest portion of its dissolved constituents being thereby precipitated. They are deposited in a mud-collector (fitted at the side of the furnace) which the water traverses on its way to the boiler proper and are blown out from time to time. Experience hitherto gained with these boilers connected to compound engines has shewn that with this arrangement, every

six months is often enough to clean them, even if it is necessary then, the tubes most exposed to the fire having only a mm or so of scale upon them although the supplementary feed is taken from the sea.

19) With multiple-expansion engines the boilers are fed exclusively with distilled water, to which lime-water is continually added. For preparing this a special tank is usually fitted above the air-pumps. A small quantity of water is fed into the top of this tank through a branch pipe of fine bore taken from the main feed delivery pipe, and another pipe from the lower part of the tank conducts the lime-water into the condenser bottom. The method observed in mixing the lime-water is to put a weight of lime into the tank every day, which at the beginning of the voyage equals the weight of cylinder-oil used from day to day, the proportion being gradually increased to three times the weight of cylinder-oil at the end of the run. The object of the lime is to neutralize the fatty acids in the feed-water due to the cylinder-oil and to take up such lime salts as remain dissolved, in spite of the heating of the water in the steam chest, to form in the mud-collector a muddy insoluble precipitate in which are also incorporated the insoluble lime-soap formed by the excess of grease as well as the uncombined calcic hydrate. The resulting water which reaches the boiler then contains only soluble salts besides the still dissolved portion of the calcic hydrate, causing an alkaline reaction. **Separation of grease.**

20) Every three months the boilers are boiled out with caustic soda. About 16 kg of this are put into each boiler and it is then *completely* filled up with fresh water. The fires are lighted and a pressure of about an atmosphere kept up for 2 or 3 hours, upon which the boiler is blown down to the ordinary water-level. The engine is then run disconnected for about 3 to 4 hours to wash out not only the boiler but also the steam-pipes and cylinders, the water carried over being quite sufficient for the latter purpose. Finally the water is completely blown out, all doors taken off, all deposit removed, and the boiler thoroughly rinsed out. The dirt found inside is never considerable. After cleaning, the tubes are examined for defects, as corrosion, blisters, or bending. **Thorough cleaning.**

21) It was at first customary, as with nearly all water-tube boilers, to remove the soot and light ash from BELLEVILLE boilers by means of a steam jet, but it was soon found that this method consumed more steam than could be made up by ordinary evaporators as fitted with triple engines. The present practice on long voyage mail-steamers is to fit an extremely liberal **Removal of soot.**

allowance of boilers and to clean three or four of them at a time during the passage. The fires having burnt down, the tubes are swept from the front with brushes, the soot pulled out, and the boilers started again. The whole of the boilers are thus treated in rotation, each one being swept about every 3 days.

Opinion on Belleville boilers.

22) Worked with all the care and attention necessary for the elaborate system described above, BELLEVILLE boilers have stood well at sea when not pushed, but as steam-producers their results are not eminently good. What they save in weight of water compared with other water-tube boilers is almost made up for by the weight of masonry for the furnaces, as well as of the feed, air-compressing, and steam-drying arrangements. At the trial-trip of the Messageries Maritimes Co's. "Australien" only 19.2 *IP* were obtained with 1 ton weight of boiler while in the still uncompleted cruiser "Bugeaud" 9000 *IP* are expected with 365 tons weight of boilers or 24.7 *IP* per ton, whereas Scotch boilers produce 18 *IP* per ton with natural and 20 to 25 *IP* with forced draught. Whether better results will be got with the BELLEVILLE boilers of the large British cruisers is a matter of speculation, and judging from the experience with "Sharpshooter" where only 21.1 *IP* with natural and 26.1 *IP* per ton with forced draught were obtained, it is not safe to predict.

Description.

23) c. **The Herreshoff Boiler of 1888**, first fitted to the steam-yacht "Jersey Lily" is shewn in Figs. 1 and 2, Pl. 33.*) It consists of a number of elements each having 5 horizontal tubes screwed with right and left handed threads into the head-pieces of malleable cast-iron. The elements are united in pairs at their lower ends by a breeches-pipe which connects them to a larger pipe placed beneath the grate and serving as a mud-catcher. The elements are similarly connected at the top to a pipe of the same size as the bottom one and carried across the boiler front as a steam-collector. Twenty-two elements make up the boiler. The lowest tubes are 33 cm above the grate. On each side of the grate there is a row of tubes similar to the elements and connected to the steam-chest by a long pipe as well as by a short branch to the feed-pipe near the mud-catcher. These side tubes, together with the long tubes connecting the elements to the mud-catcher, are intended to prevent too great radiation from the fire to the boiler casing. Above all the tubes composing the elements there are three horizontal rows of tubes within the casing, connected together into one continuous tube

*) Annual report of the chief of the bureau of steam engineering for the year 1888. Washington 1888. p. 48.

in which the feed-water is warmed up and which also serves to protect the roof of the boiler casing against radiation from the hot gases. At the upper part of the front is placed the trap or water separator from the top of which the steam-pipe is led to the engine, the trap being connected at about the middle of its height with the steam-collector. The feed-pipe coming from the feed-heater is introduced into the bottom of the trap and just above the bottom a pipe branches off which carries the heated feed-water into the mud-catcher by gravitation. The casing consists of two thicknesses of 1.5 mm sheet-iron with a 30 mm space between them filled with slag-wool. The ash-pit plates are 3 mm thick. The whole of the boiler and feed-heater tubes are of 25 mm internal diar. and are separately tested to 70 atmos. A larger boiler subsequently built for the steam yacht "Say When" has the steam-collector inside the casing and the trap at the boiler back. The elements have each 7 separate tubes of 50 mm internal diameter, the breeches-pipes to connect the vertical rows of tubes with the mud-catcher are omitted, each row being separately jointed to it by a branch-pipe. These branches are set in a zigzag line on the mud-catcher. The connection is similar to that in the PERKINS boiler with right and left-handed threads and sockets. The joints are made with asbestos washers. The boiler casing is not double but the single thickness of sheet-iron is provided with a series of T-iron stiffeness 63.5 mm deep which serve to hold a lining of magnesia fire-bricks.

24) The "Jersey Lily's" boiler was tested by a U. S. Government Commission. With a combustion of only 45 kg per sq. m of grate per hour 1 kg of good anthracite coal converted 6.37 kg of water at 16.7 ^{0} C into steam of 8.46 atmos. working pressure, and 3.5 % of moisture. The weight, including fittings, of the empty boiler was 1335.8 kg, that of the water 47.6 kg or only about $\frac{1}{28}$ of the weight of the boiler. Nothing further is known as to its performance nor that of the "Say When", which leads to the presumption that neither was remarkable, otherwise the torpedo-boat "Cushing" completed in 1891 (compare § 47, 18) would certainly have had these boilers and not THORNYCROFT'S. The HERRESHOFF, like the BELLEVILLE boiler, requires a special feed-pump, but as its arrangements for uninterrupted working are not so ingeniously and completely developed as those of the latter, it must be considered inferior. **Opinion upon the Herreshoff boiler.**

25) d. **The Alban Boiler of 1843** *) is probably the earliest practicable **Description.**

*) E. ALBAN. Die Hochdruckdampfmaschine. Rostock 1843. p. 214.

water-tube boiler that remained at work, although BURGH*) mentions a water-tube boiler with vertical tubes patented in England in 1824, invented by Moore, an amateur, from the drawing of which it appears doubtful whether it could ever have been put together with the tools and appliances available at the time. Others of the older English water-tube boilers, as those of HANCOCK, OGLE, SUMMERS, DANCE, and MACERONE were unsuccessful because they had neither water-separator nor steam-collector. The ALBAN boiler, although only having a water-chamber at the front end, must be regarded as the progenitor of all the water-tube boilers with water-chambers described here. Its tubes were closed at the back end and not arranged in vertical rows as in the elements of the modern boilers, but fixed into a tube-plate in diagonal rows in order to better absorb the heat of the combustion gases. With the object of keeping up something approaching to circulation in these boilers the mouths of the tubes (which were of copper) communicating with the water-chamber at the front, were not quite open, but for each tube two oval openings were cut in the tube-plate, the water entering the tube through the lower and the steam issuing through the upper. Strips were fitted so as to conduct the mixture of steam and water coming from the tubes to the left side of the water-chamber whence it rose to the trap above. This was connected by an upper and lower pipe with the steam-collector which was placed on the same level and from which the steam was finally taken. To the bottom of the steam-collector a pipe led down on the right-hand side of the water-chamber and carried back the water separated out by the trap down below the bottom row of tubes. The feed-water entered the trap. Many of these boilers were built by ALBAN in his time.

Difference between boilers with elements and those with water-chambers.

26) Boilers with elements, as distinguished from those with water-chambers, have the advantage of greater elasticity, they can withstand more sudden heating and cooling without becoming leaky, because their tubes can expand more freely than those of the water-chamber boilers which are rigidly confined between the tube-plates. On the other hand the water-chamber boilers can with impunity be more severely forced in consequence of their better circulation and are also more rapid steam-producers. In recent water-tube boilers with 2 water-chambers, the feed-water having been warmed up in the bottom rows of tubes (which are placed with a rising angle of 10^0 to 15^0), flows

**) J. P. BURGH. A practical treatise on boilers and boiler-making. London 1881. p. 5.

into the front water-chamber and thence through the steam-collector or through the upper tubes where the steam-space begins, into the back water-chamber. The tube-plates require no staying beyond that afforded by the tubes which are expanded as usual and by the steam-collector which is riveted to both water-chambers.

27) **e. The Heine boiler of 1887*)** does not differ very essentially from ALBAN'S. The chief improvements it embodies are that it has two water-chambers, one at each end of the tubes, these being expanded into the tube-plates, and that the chambers are fitted with hollow screwed stays through which a steam jet can be applied for removing soot from the tubes and which possess the advantage of immediately giving notice of a fracture by the escaping steam or water. HEINE further attempted to more effectually separate the water from the steam by fitting baffles over the mouths of the water-chambers in the steam-collector, also to use up more completely the heat of the furnace-gases by means of fire clay slabs laid on the tubes, causing the gases to make a somewhat circuitous path instead of a direct one to the chimney. The other German boilers with two water-chambers, as STEINMÜLLER'S of Gummersbach and BÜTTNER'S of Ueberdingen, are very similar to HEINE'S and possess many good qualities, but have not been applied to marine purposes. **Improvements**

28) A boiler allied to HEINE'S, but better adapted for sea service was fitted in 1890 to a tug called "Föhn" at the Imperial Dockyard Kiel, as shewn in Figs. 1 and 2, Pl. 31. While HEINE'S tubes slope upwards from the front, in this boiler and in all similar marine boilers they have the opposite inclination. The steam-collector is smaller than HEINE'S, but on the other hand the number of tubes considerably greater and the steam-collector is kept closer down to the tubes, to save height. The boiler has two furnaces bounded on the outside by two vertical rows of tubes and separated in the middle by one. The bottom horizontal row of tubes is 500 mm above the grate and covered at both sides, as shewn in Fig. 1, rather beyond the middle of the furnace with firebrick, so as to form, as in the d'ALLEST boiler, two combustion-chambers which only allow the gases to ascend in the middle between the tubes. The water-chambers also have hollow stays permitting the tubes to be cleared of soot through them. The tube ends are jointed into the water-chambers with cast steel ferrules and asbestos washers (Fig. 1, Pl. 39) **Boiler of the tug "Föhn."**

*) Deutsche Patentschrift. Nr. 751 of Aug. 15, 1877.

introduced through two oval openings in the bottom of the boiler. The steam-collector has a baffle plate above the mouth of the front water-chamber and under the steam-pipe. The staying of the steam-collector at the mouths of the water-chambers is shewn in the figures.

Opinion on Heine's boiler. 29) "Fōhn's" boiler weighed 8544.5 kg complete and the water 2590, or 11134.5 kg together. Her compound engine developed 234 *IHP* on trial with 7.86 atmos. working pressure. She burnt 100 kg of coal per sq. m of grate and 1.067 kg per *IHP* per hour, producing 21 *IHP* per ton of boiler and 93 *IHP* per sq. m of grate, or nearly as much as BELLEVILLE boilers, but at this rate the boiler was forced to its utmost. It requires to be worked very gently under ordinary conditions, only 60 kg of coal being burnt per sq. m of grate, the revolutions come down from 181 on the trial-trip to about 120, and less horse-power per ton of boiler is produced than in Scotch boilers.

Description. 30) f. **The Oriolle boiler of 1885** *) is closely allied to HEINE'S as is obvious from Figs. 3 and 4, Pl. 31. Its water-chambers have the disadvantage of parallel sides, while those of "Fōhn's" boiler diverge upwards in proportion to the steam generated in the tubes. The steam-collector is not placed within the boiler casing, but at the side of it and is common to two adjacent boilers in one stokehole. The staying and jointing of the water-chambers is similar to HEINE'S. There is only one furnace to each boiler and the combustion-gases pass direct to the chimney. As in BELLEVILLE'S only the lower rows of tubes are filled with water which renders the boiler very light, but as there is neither feed regulator nor steam-drier, the working pressure is subject to great variations, unless the firing is extremely gentle. That not so many French torpedo-boats are fitted with them as formerly is a proof that they were not particularly successful. In 1880 ORIOLLE fitted a steamer named "Mitidjah" with his boilers but nothing has been reported as to her results.

Description. 31) g. **The D'Allest boiler** **) **of 1890** is shewn in Figs. 8 and 9, Pl. 31. Differing from the hitherto described boilers of this type the tubes are arranged horizontally instead of diagonally. The spaces between the lower tubes are completely closed with fire-clay lumps so as to entirely surround the furnaces which are roomy. The boilers are placed side by side in pairs and the combustion-gases leave the furnaces towards the centre, passing

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*) Transactions of the Institution of Naval Architects. London 1894. p. 93.

**) Journal of the American society of naval engineers. 1895. p. 386.

thence among the tubes on both sides and are then led through an uptake along the side of the boiler under the steam-collector up into the funnel, the top row of tubes being likewise blocked off with fire-clay lumps. In order to sufficiently absorb the heat of the gases; the spaces between the lateral tubes are closed down to the bottom third by iron plates. The arrows in Fig. 8 indicate the course of the gases. In consequence of these arrangements the combustion of the D'ALLEST boiler is perceptibly better than in any other water-tube boiler, as proved by all trial-trip results hitherto obtained. In order to provide against the sagging of the tubes mentioned under BELLEVILLE boilers, D'ALLEST uses Serve tubes in the bottom rows and in several other positions for the sake of the stiffness afforded by their internal ribs — of course perfectly pure feed-water is then a sine quâ non. For the better securing of the tubes in the tube-plates a ring of thin wire is placed inside the tube end before inserting the expander. The hollow stays with covers on their ends jointed with asbestos cloth treated with india rubber are the same as those of other boilers of this type. The water-line is kept a little above the bottom of the steam-collector. Before the tubes are put in, the tube-holes are jointed with blank flanges and the steam-collector and water-chambers submitted to a water pressure of 22 atmos. Recently in some cases the tube-holes, hand-holes, &c. have been bored after this water-test, to make sure that there is no deformation of any of the surfaces through imperfect bearing of rivets or other connections.

32) Six-hour and three-hour evaporative trials were carried out by the French Admiralty with the D'ALLEST boilers of the ironclad "Carnot", the feed water temperature being 21^0 to 25^0 C and the working pressure 3 to 3.5 atmos. While 50 kg of good Cardiff coal were burnt per sq. m of grate per hour the factor of evaporation was 10.67, which however fell to 8.75 when the rate of combustion was raised to 150 kg. The proper working pressure of these boilers was 15 atmos. and each weighed complete and empty 11742 kg, the weight of water being 3107 kg. Similar results were obtained with the coast-defence ship "Bouvines", the factor of evaporation being 10 at 60 kg burnt per sq. m of grate per hour and 9.2 at 150 kg. The boilers of the cruiser "Chasseloup-Laubat" weigh 335 tons and the water 53 tons or about a sixth of the boilers. **Trials.**

33) The D'ALLEST boiler requires to be as gradually warmed up as a Scotch boiler. As with BELLEVILLE'S, lime-water is mixed with the feed in the proportion of about 3 kg per 1000 *IHP* per 24 hours, when the boilers are filled with fresh water and **Working.**

distilled auxiliary feed is used. Every 8 hours the boilers are well blown and scummed. No salt is allowed in them, though BELLEVILLE does not object to a certain amount of it. If, in consequence of a leaky condenser the boilers salt up at all, the density must not be allowed to exceed 2, or at the outside 4 per cent of salt, but some of the boilers must be shut down and the others forced in order to increase their circulation and diminish the chance of deposits being formed. Like BELLEVILLE'S, these boilers are washed out from time to time and for this purpose D'ALLEST introduces caustic soda in the proportion of 5.5 kg to 1 ton of water in the boilers, but to save the pumps from injury, the solution is made as weak as 1 kg of soda to 71 kg of water. The boiler is then boiled out for 2 or 3 hours under a pressure of 2 to 2.5 atmos. and afterwards completely blown out. The tubes and steam-collector are wiped out. — The only difficulty is the removal of the soot when under steam, it is done by means of a steam-jet, brushing, and shovelling. The steam is applied for three minutes at a time which causes the water-level to fall about 10 cm.

Opinion on the D'Allest boiler.

34) According to existing experience the D'ALLEST boiler is the most successful of all the straight tube water-tube boilers. It is however very sensitive to rapid warming up, does not require such complicated arrangements for working as BELLEVILLE'S, but must be very uniformly fired or violent oscillations in the pressure will occur. It may be remarked that it contains a rather large quantity of water. At the forced trials of the iron-clad "Jemappes" with 9250 *IHP* 0.921 kg of coal were burnt per horse per hour at a rate of combustion of 145 kg per sq. m of grate. On easing down to 107.6 kg per sq. m with 7711 *IHP* only 0.82 kg per horse per hour were burnt. These consumptions are higher than those of Scotch boilers under the same conditions. The cruiser "Chasseloup-Laubat" on her forced trials attained 9700 *IHP* with 335 tons of boilers, or 29 *IHP* per ton which excels any performance of Scotch boilers hitherto.

Description.

35) h. **The Yarrow boiler of 1892** *) (Figs. 5 and 6, Pl. 31) differs from the HEINE and ORIOLLE boiler only in having no screwed stays between the flat sides of the water-chambers. The tubes are expanded into these four plates and where they pass through the space between the plates they are provided with slits on the top and underneath, their outer ends being closed with screwed plugs. In order to baffle the gases on their way to the funnel, plates are laid upon the tubes, or these are provided

*) English Patent. No. 18965 of 12. 10. 93.

with external ribs (similar to the internal ones of Serve tubes). These boilers have all the drawbacks of ORIOLLE'S inasmuch as they contain very little water. Several of them were fitted to British torpedo-boats, but it does not appear that they were particularly successful, at any rate nothing has been published about them as would otherwise have certainly been the case.

36) i. **The Mosher Boiler of 1895,***) which strictly speaking does not belong to this group, is shewn in Figs. 8 to 11, Pl. 33. Each of its two water-chambers carries a steam-collector which is connected to the boiler-casing. The tubes are not straight but curved; they rise from the water-chamber on each side, are carried slanting somewhat upwards over to the opposite side, and are then returned to the steam-collector over the water-chamber from which they started. A similar course is followed by the tubes shewn dotted in Fig. 8, this single row forming the back end of the boiler. The boiler thus consists of two separate parts, the tubes of which cross each other, enclosed in one casing, but connected only through the steam and feed-pipes. In this way the boiler is rendered as elastic as those without water-chambers. There is a very roomy furnace fired by three doors. The tube clusters are separated by baffle plates which direct the gases towards the steam-collector. The tubes are of 25 mm external diameter and are closed at their lower ends by screwed plugs as shewn in Fig. 8. To save staying the bottom of the steam-collector, the tubes being weakened (as stays) by the slits through which the communication with the water-chamber is established, as well as to compensate for the loss of strength due to the riveting (Fig. 11), this bottom plate is made 13 mm thick, the rest of the steam-collector plating being only 8 mm. This boiler is designed for 1000 *IHP*, corresponding to 270 *IHP* per sq. m of grate per hour and it is questionable whether it will come up to such expectations. To a certain extent it is related to the TOWNE boiler (see § 59) which has not quite answered to the requirements of actual work, but as a superficial comparison shews, it embodies considerable improvements upon the latter boiler. Description.

37) k. **The Babcock-Wilcox boiler of 1893****) belongs to a group of water-tube boilers forming the connecting link between the water-chamber and the element boilers. Its arrangement is precisely similar to that of the boilers with water-chambers, but of these it has a considerable number, in place of two. As in the BELLEVILLE boiler of 1878, each pair of tubes were united to Description.

*) American Engineer. New York 1895. p. 466.

**) Transactions of the Institution of Naval Architects. London 1894. Pl. XXIV, pp. 89 and 329.

form one element, here there are two or four tubes connected in a cluster placed two and two diagonally right and left above each other, these clusters being then vertically superimposed. Each such series of clusters possesses at each end a separate water-chamber or "header" the sides of which have a wavy form in front elevation, in order to get the diagonally placed pairs of tubes close together like NICLAUSSE'S Fig. 12, Pl. 30. Sometimes however the tubes belonging to each pair of headers are placed in one vertical row. From each of these water-chambers a pipe leads to a common steam-drum which is half filled with water. The tubes coming from the front headers are connected to the drum at about the water-level and carry up steam. The unevaporated water passes back into the tube clusters by the connecting pipes running down from the drum to the back headers. Besides the sloping tubes there is a single layer of closely-spaced vertical tubes placed at each side of the boiler like a casing. These tubes open at top and bottom into two rectangular water-boxes, the upper one of these on each side being connected by a horizontal pipe with the water-space of the drum. The circulation is produced by the water from the drum flowing down by three or four of the back tubes into the bottom rectangular water-boxes and rising again by the side tubes. The steam-collector is placed above the drum and joined to it by two necks and in the funnel which is in front of it is a voluminous feed-heater. The tube-end doors in the headers are contrived so as to be tight without packing. The tubes are expanded into the headers which are built up of steel plates welded together and pressed into the waved form. In general the boiler may be regarded as an improved form specially adapted to marine purposes of the inventor's land boiler exhibited in 1889 by the Berliner Maschinenbau-Act-Gesell*) at the Accidents Prevention Exhibition in Berlin.

Opinion on the Babcock-Wilcox boiler.

38) BABCOCK-WILCOX boilers, unlike the other water-tube boilers hitherto described, are made as large as ordinary marine boilers with 3 and 4 furnaces, which can be simply done by putting in more headers and tubes. Several have recently been fitted to cargo steamers with 14 to 15 atmos. working pressure for quadruple engines, particularly by builders on the East Coast of England. They contain 5 tons and upwards of water, so that HOWDEN'S statement that the cargo steamer "Nero" only

*) Zeitschrift des Vereins deutscher Ingenieure 1889. P. 672.

produced 10.8 *IHP* at sea per ton weight of boiler is probably near the truth. The manufacturers of these boilers report differently; they say that in the American yacht "Reverie" a factor of evaporation of 9.26 was obtained on an 8 hours' trial with 14.4 atmos. working pressure and a combustion of 68 kg per sq. m of grate per hour — very moderate firing. On the other hand, one of these boilers tested in America by Naval Engineers*) only reached a factor of evaporation of 7 to 7.5 during a 24 hours' trial with a combustion of 200 to 215 kg per sq. m of grate. This boiler weighed 22.34 tons with water and the water alone 4.28 tons or about $\frac{1}{5}$ of the total. Some English yachts have also been fitted with these boilers but nothing has been published as to their weights and performances. They have plugs instead of doors opposite the tube ends in the headers as in the American torpedo-boat boiler Fig. 10, Pl. 13, — an innovation which is now particularly favoured in America. Until the BABCOCK-WILCOX boiler is proved to produce as the manufacturers claim, 77 *IHP* per ton of weight, instead of only 20.6 as in the lake steamer "Zenith City"**) the boilers of which weigh 78.87 tons and develop 1826 *IHP*, it will have as little chance of becoming extensively adopted in war-ships as its rivals. But the extraordinary energy displayed by the manufacturers and the results they have recently obtained with cargo steamers may probably soon render these boilers more popular in the mercantile marine.

39) **The Dürr boiler of 1883*****) belongs to quite a special class of water-tube boilers comprising but few besides itself and the NICLAUSSE boiler. In the DÜRR boiler, obviously copied from the latter, a well-known arrangement, that of the FIELD tube is applied with much skill (Figs. 13 and 14, Pl. 30). The boiler has only one water-chamber (at the front) which is welded together and stayed in the ordinary way, the tubes are closed at their back ends and rest in an iron wall lined with fire-brick. To this wall a wrought-iron frame is secured which supports the top drum The back ends of the tubes are closed with hollow cones ground in and screwed up from outside with bolts (Figs. 5 to 7, Pl. 39). The cones are provided with flanges to prevent their being drawn into the tubes after protracted use and their hollow form gives them the necessary elasticity. The tubes, unlike those of all Description.

*) Journal of the American Society of Naval Engineers. Washington. 1895. p. 694.

**) Journal of the American Society of Naval Engineers. Washington. 1895. p. 765.

***) C. BUSLEY. Die jüngsten Beobachtungen und Erfolge des deutschen Schiffbaues. Berlin 1895. P. 41.

boilers hitherto brought out, are inserted obliquely and not square into the tube-plates and are provided with rings welded on (Fig. 6, Pl. 39) and turned up conical in special lathes, the holes in the tube plates being bored to fit them. DÜRR employs this very costly arrangement to avoid giving his water-chambers an inclined position, so that they may be better supported on the boiler seating and also to keep the hot gases from directly impinging on the joints of the tubes in the water-chamber. The water-chamber is divided into two parts by a plate which receives the somewhat bell-mouthed ends of the internal tubes. The lids opposite the tube-ends are of a light hollow section, jointed metal to metal, and arranged to keep tight even if the openings lose their circular form through distortion of the water-chamber (Figs. 5 and 6, Pl. 39). The water-chamber encloses the steam-collector and the tubes above the latter act as steam driers. To obtain a regular flow of steam, the upper space in the chamber is divided similarly to the lower one and the steam-drying tubes are arranged like the water-tubes. The steam is taken from the back compartment of the upper chamber. The furnace is bounded on each side by a wall of tubes which are bent alternately right and left so as to be efficiently connected to the water-chamber. The circulation takes place by the heated water flowing off from the tubes into the back compartment of the water-chamber, whence the steam ascends to the steam-collector. The unevaporated water flows from the latter back into the front compartment of the water-chamber and thence through the internal tubes into the water-tubes again. The soot is to be removed by sweeping with the assistence of the steam jet.

Opinion on the Dürr boiler.

40) With certain improvements in detail specially directed to sea service, the DÜRR boiler has prospects of extended use. Among these developments would be the placing of the tubes square to the tube-plate and expanding them into it to diminish the risk of their drawing out, and in combination herewith, an alteration of the back end plugs, rendering them removable from the outside, like for instance BÜTTNER'S arrangement shewn in Figs. 8 to 10, Pl. 39. With these and similar improvements, it is hoped in the German Navy, where extensive trials of the DÜRR boiler have been made, to obtain better results than in the small transport steamer "Rhein". This vessel's boiler of 14 atmos. W.P. weighs 21.98 tons, of which 4.1 tons or about $\frac{1}{5}$ of the weight of the boiler, represent the water, so that there is fully three times as much water as in BELLEVILLE'S

boiler. At a six hour's forced trial in May 1894 the boiler certainly worked faultlessly but only developed 350 *IHP* in the triple-expansion engine when burning 100 kg on 9 sq. m of grate per hour or not quite 16.5 *IHP* per ton weight of boiler, the coal-consumption being 1 kg per horse per hour. Since then the DÜRR boiler has been considerably improved. The diameter of the tubes has been reduced from 114 to 83 mm and the pitch from 250 to 174 mm. One of these marine boilers for an engine of about 800 *IHP* with 5.03 sq. m grate and 223 sq. m heating surface weighed 20.76 tons and the water 4.6 tons. The evaporation was 7.88 kg of water from 0^0 C per kg of coal at a combustion of 200 kg per sq. m of grate per hour. The *IHP* per ton of boiler is about 30 which compares quite favourably with the French boilers.

41) m. **The Niclausse Boiler of 1892***) forms in general a combination of the DÜRR and BABCOCK-WILCOX boilers. With the former it has the internal suspended tubes and with the latter the waved form of water-chamber in common, as shewn in Figs. 11 to 12, Pl. 30. All the water-chambers are connected to a steam-collector having a dome in its centre. The material of the water-chambers is malleable cast-iron, a division plate separates them vertically into 2 halves, the back compartment forming the path of the heated water and steam to the steam-collector, and the front one conducting the unevaporated water back again. For each tube an oblique hole is made through the front and back of the water-chamber as well as through the division-plate, those in the water-chamber are bored conical, the large diameter of the back hole being exactly equal to the small diameter of the front one. As in DÜRR'S boiler the tubes are placed on a slant and reduced in diameter at the back end and closed with cap-nuts of such diameter as to admit of the tubes being drawn out in front. The front ends of the tubes are confined in their places by certain fittings of malleable cast iron, called by the inventor "lanterns" (Figs. 11 to 17, Pl. 39). The back ends of the lanterns which fit into the back tube-plate are strong and heavy, the front ends are lighter to give more elasticity. All the lanterns are made to template so as to be interchangeable. The middle of the lantern fits exactly into the division plate and on each side of this part of the lantern the barrel is cut away to assist the circulation of the water (Fig. 14). The tubes are screwed on to the thin ends of the lanterns (Fig. 13) so that the joint comes exactly in the Description.

*) Journal of the American Society of Naval Engineers. Washington. 1895. p. 402.

centre plane of the division-plate, the cone assisting the joint. The suspended tube in each water-tube is kept in its place by a second lantern which only extends from the front plate to the division plate and is screwed in front into the big lantern. The suspended tubes are of thin sheet, put together in two halves (Fig. 18, Pl. 39) on special machines, riveted into the lanterns, and beaded over (Fig. 16). The water-tubes are arranged in pairs held together by a strap and can be taken out and changed so simply that when the boiler is cold and empty a new tube can be put in in two minutes. The connecting pipe from each water-chamber to the steam-collector is bell-mouthed at the upper end in order to lower the velocity of the issuing steam and give it time to dry. The water-chambers are all connected to one pipe at the bottom, so that the boiler can be blown out. The feed-water enters a special receptacle in the steam-collector, which retains the impurities and has to be frequently cleaned. From this receptacle the water passes down into the front compartments of the water-chambers. The steam-pipe starts from the dome.

Opinion on the Niclausse boiler. 42) In face of the circumstance that the principal German firms manufacturing malleable cast iron will not guarantee it for a tensile strength of 36 kg per sq. mm and that there is a strong tendency to exclude cast iron altogether as a material in boiler making with present pressures, the arrangement of the NICLAUSSE boiler with its water-chambers and lanterns cannot be considered a particularly happy one, although it must be admitted that the malleable castings used have shewn excellent bending and hammer tests. The amount of machine work on the lanterns also makes the boiler very expensive. On the other hand it can, like the DÜRR boiler, stand severe forcing as compared with the water-chamber boilers, although the lower rows of tubes are then stated to begin to sag like BELLEVILLE'S which may perhaps be got over by the use of Serve tubes as in the D'ALLEST boiler. The suspended tubes are not supported as they are constantly vibrating while at work. The NICLAUSSE boiler is subject to violent oscillations of pressure when irregularly fired, in consequence of its small weight of water, but it can be worked like a Scotch boiler without BELLEVILLE'S feed and steam-drying arrangements. One particular advantage is that all difficulties can be dealt with from the boiler-front and the tubes cleared of soot with a steam-jet inserted through the interstices of the water-chambers from the same position, beginning at the top rows. When these boilers are fed with distilled water they do not require any special attention, it is

sufficient to blow and scum them once a day. If the tubes want cleaning the best way is to take them out and this opportunity can be used to wash out the water-chambers. NICLAUSSE boilers are fitted to the French dispatch boat "Elan", the cruiser "Friant", and the Russian gun-boat "Khrabry". Up to now (March, 1895) "Friant" has only completed her trials, developing 9438 *IHP* with a combustion of 122 kg per sq. m. The consumption was 0.911 kg per horse per hour. At a combustion of 89 kg per sq. m 7189 *IHP* were developed at 0.859 kg per horse per hour or rather more than with cylindrical boilers. As NICLAUSSE*) states the weight of the boilers to be 318.94 tons, they produced 29.6 *IHP* per ton, thus rather excelling the highest performance of Scotch boilers. The weight of water was 47 tons or $\frac{1}{7}$ of the total against $\frac{1}{17}$ to $\frac{1}{19}$ in BELLEVILLE'S. Elaborate evaporative trials of NICLAUSSE boilers were held by KENNEDY and UNWIN at THAMES DITTON in the spring of 1894, when an evaporative factor of 8.68 was obtained at a combustion of 176 kg and an average of 155 *IHP* per sq. m of grate per hour.

43) **III. Water-tube boilers with curved tubes** are distinguished before all other water-tube boilers by great elasticity which admits of their being rapidly warmed up. They all have either one or two top and bottom drums connected by variously curved tubes. They contain very little water which is always in a state of vigorous circulation. The boilers of this group have achieved trial trip results never before reached by any boilers at all. But with all of them there is more or less difficulty in the way of inspection, cleaning, and the renewal of defective tubes. **General.**

44) n. **The Du Temple boiler of 1886** was first tried in the French torpedo-boat No 20**) and is shewn in Figs. 1 and 2, Pl. 32. It is composed of a top drum placed in the centre line having a dome and two bottom drums at the sides connected by the tubes to the top drum which is half filled with water. The tubes are of steel 25 mm external diameter, each four composing an approximately parallel group. They are all bent worm-shaped, expanded into the bottom drums, and jointed into the top drum with internal nuts to keep them from drawing. At the front a down-comer leads from the collector to each of the bottom drums. The furnace is surrounded with fire-brick **Description.**

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*) Générateur inexplosible, BREVÊT, NICLAUSSE. Paris 1896.

**) Revue industrielle of Jan. 5, 1889. p. 5.

lumps attached to the casing of sheet steel. The casing encloses the top drum and carries the funnel on its top. Owing to the small weight of water and its ingenious distribution over the very large heating surface, the generation of steam is pretty rapid. Steam has been got up in a 500 horse-power boiler in 45 minutes. The circulation is very active, owing to the vigorous evaporation in the serpentine tubes. The water carried up from these into the top drum returns through the drown-comers at the front into the bottom drums to which the feed-pipes are connected. At the back end of each of the drums there is a blow-off pipe, the three pipes being connected to one cock, and as they originate in those places where the water is quietest and therefore any impurities have a chance to settle, these can be easily got rid of. Scale is the less to be feared as only fresh feed is used. There are no special feed arrangements as in other boilers of this group.

Opinion on the Du Temple boiler.

45) The two DU TEMPLE boilers of torpedo-boat No 20 have made 27 trips in their first two years, 20 of which were at high speed, without giving any particular trouble. A number of French torpedo-boats were afterwards fitted with them. It is asserted in French technical journals that a 500 horse-power boiler only weighs 5.5 tons but in view of the highest results got out of the most recent boilers of the class, this must be doubted. As regards durability and facility of repairs, it is reported that the tubes do not last on account of their great length and the design has consequently been modified so as to resemble more closely that of the NORMAND boiler.

Genesis.

46) o. **The Thornycroft boiler of 1887 and 1889.** Having constructed a spiral boiler very much like HERRESHOFF'S in 1882 for a small shallow-draught steamer intended for mission service in Central Africa (see § 59), the drawing of which afterwards appeared in Engineering*), THORNYCROFT in 1886 introduced a novel water-tube boiler for a British second class torpedo boat, from which the two boilers of the Spanish torpedo-boat "Ariete"**) tried in 1887 originated. Figs. 3 to 6 Pl. 32***) shew the boiler of the British torpedo-boat referred to, which corresponds in design with the "Ariete's" boilers, only differing from them in size, fittings, and mode of placing in the boat.

Description.

47) The THORNYCROFT boiler has a horizontal cylindrical steam-collector at the top and at the bottom one horizontal water-

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*) Engineering 1883. I. p. 463.

**) Zeitschrift des Vereins deutscher Ingenieure. 1887. p. 911.

***) Engineering 1887. II. p. 104.

cylinder on each side of the furnace, two comparatively large descending pipes or down-comers connecting the top and bottom drums, and a large number of small water-tubes. As in most recent water-tube boilers the feed-water enters at the bottom of the top drum (see longitudinal section Fig. 4, Pl. 32), flows down the descending pipes into the bottom drums and through these into the tubes where it is converted into steam which passes off at the top of the top drum. The only difference between the boiler shewn and the "Ariete's" is in the arrangement of the descending pipes which in her were at the front to save room and are here at the back. The average water-level is at about half the depth of the top drum, but as „Ariete's" trials proved, its position is not important.

48) In order that the top drum may perform its office of separating the steam from the priming water as efficiently as possible, it must be withdrawn from the direct influence of the hot gases, or the active ebullition would cause two much agitation of the surface of the water. With this object the upper portions of the water-tubes are curved round the drum, as shewn in Fig. 3, Pl. 32. As the inner rows of the tubes on each side approach each other very closely in the centre, they form a sort of screen over the furnace and protect the drum from the flame, in which they were assisted in the early boilers by a layer of asbestos, afterwards omitted. **Top drum.**

49) The tubes are of steel, arranged in zigzag rows, and expanded into the top and bottom drums in the usual manner. The two outside tubes in each row are bent near their ends so as to touch each other and form a closed screen to prevent the escape of the furnace gases. The inner tubes preserve their zigzag order up to the top drum and permit the gases to pass through their interstices. The number of inner tubes can be varied at will and only depends upon the available shell area of the drums. The gases, ascending among the tubes, follow the course of the latter partially round the top drum and pass up the funnel above it. A casing of thin sheet surrounds the tubes, lined on the inside with fire-clay and on the outside with asbestos. **Arrangement of the tubes.**

50) In order to get as much combustion space as possible, the 1887 boiler has only the 8 front and the 8 back rows of tubes bent as shewn on the left side of Fig. 3, all the other tubes having the form exhibited on the right hand side of the figure. The front and back groups are intended to form a kind of boundary to the furnace. The open space between the two series (right and left) of tubes is occupied at the front by the fire-door and **Furnace.**

at the back is filled up with fire-brick, also employed to cover those portions of the bottom drums which would otherwise be exposed to the direct action of the flame.

Circulation. 51) Whereas boilers with straight tubes, both horizontal and inclined, suffer from priming, THORNYCROFT'S boiler is said to be actually favoured by the very conditions which elsewhere give rise to this trouble. The more rapid the ebullition in the tubes, the greater the velocity with which the water rushes up into the top drum and down the down-comers and therefore the better the circulation. Although the nascent steam exerts the same pressure downwards on the water in the bottom drums and upwards on that in the top drum, the difference in density between the mixture of steam and water in the tubes and the solid water in the down-comers is sufficient to maintain an uninterrupted and very vigorous circulation. As boilers with particularly good circulation are inclined to give wet steam, Mr. THORNYCROFT has devised a separating arrangement now to be described. A piece of sheet iron bent into the form of a half cylinder and having saw-like teeth at each side is fixed in the top drum close in front of the mouths of the water-tubes (Figs. 3 and 4) so as to receive the impact of the steam and water issuing from them. A portion of the water is at once separated out, the remainder is arrested on striking the bent up teeth of the baffle plate. The peculiar form of these teeth is the result of a long series of experiments and has shewn itself very effective for its purpose. The teeth are formed by cutting (as shewn in Fig. 5) a triangular piece out of the sheet, continuing this cut, as indicated by the full line, further into the sheet and then bending up the ends rectangularly at the places marked by the dotted lines, as in Fig. 6. The steam thus freed of water, passes up into the steam pipe which is fixed in the upper part of the drum and provided with slits in the usual manner.

Trial-trip results. 52) The contractor's trial-trip of the "Ariete" was held at the mouth of the THAMES in July 1887. She attained a mean measured-mile speed of 26 knots, and during a two hours' run, the speed, calculated from the revolutions, was 24,9 knots; the full power *IHP* was 1570.

Disadvantage. 53) A serious draw-back to the THORNYCROFT boiler is its want of accessibility for repairs. If one of the inside tubes becomes defective, all the tubes outside it in the same row must be cut out in order to replace it and of course the boiler must be empty and cold. Mr. THORNYCROFT considers however that his boiler requires but few renewals as its very perfect

circulation and great elasticity render leaks rare. Experience seems hitherto to corroborate his view, for anything like frequent trouble with the tubes has not arisen, although their upper parts have been observed to get red-hot when the water was low.

54) The form of the 1889 THORNYCROFT boiler, which is obviously allied to the DU TEMPLE boiler, differs from that described above in that the front and back tubes are bent to the same shape as the middle ones (shewn on the right hand side of Fig. 3, Pl. 32) and the furnace is thus bounded by fire-brick instead of tubes. This alteration makes the furnace more accessible and enables two fire-doors to be fitted. The descending pipes or down-comers between the top and side drums are kept at the front to save room. With this boiler Prof. KENNEDY made the evaporative trial already referred to in § 22, 60 and found that the best natural-draught result was 13.4 kg of water evaporated from and at 100° C per kg of coal (or probably more correctly pure fuel), the theoretical evaporative power of which, according to analysis, was 15.41 kg, shewing the efficiency of the boiler to be 0.87, — an unusually high figure and scarcely credible as compared with the torpedo-boat "Cushing's" results. In her trials which were conducted by LORING the highest performance gave an evaporative factor of only 10.55 at a combustion of 37 kg per sq. m of grate with 8.09 % of moisture in the steam, and at a combustion of 324 kg per sq. m of grate the evaporative factor fell to 5.3. The boiler used in these trials weighed 11 tons including 2 tons of water, so that the weight of the water was between $^1/_5$ and $^1/_6$ of that of the boiler. Mr. THORNYCROFT states that on forced trials these boilers have developed 68 *HP* per ton weight of boiler. The author has himself seen them warmed up in 15 to 20 minutes without injury. General arrangement.

55) They have been fitted not only to British, Spanish, French, and American torpedo-boats, but also to larger ships, as the British torpedo-cruiser "Speedy", the Danish cruiser "Geyser", and the German armoured vessel "Aegir". Judging from these orders, the boilers must have given satisfaction. They are especially well reported of by the Danish Admiralty*) with reference to "Geyser's" trials, although only 39 *HP* per ton of boiler were developed, whereas "Speedy" reached 52 *HP* according to Mr. THORNYCROFT'S statement**). "Geyser's" boilers weighed 108.2 tons with the water, the latter alone 17.4 or about Behaviour on trial-trips.

*) Mittheilungen a. d. Gebiete des Seewesens. Pola 1895. p. 629.

**) Transactions of the Institution of Naval Architects. London 1894. p. 331.

$^1/_6$ of the boilers, the proportion being thus about the same as NICLAUSSE'S.

Behaviour at sea. 56) But to keep up good results with recent water-tube boilers they have to receive such extreme care and attention as must, under the conditions of service of a torpedo-boat, interfere with its tactical usefulness. An instance of their great sensitiveness to careless treatment is afforded by the Spanish torpedo-boat manœuvres of 1888*), at the close of which every boat had to make a $3^1/_2$ to 4 hours' full-power run. "Rayo" and "Ariete" which had both attained 26 knots on the mile at their contractor's trial-trips in England, only got up to 19.5 and 17.2 knots respectively, though it is to be noted that "Ariete" could only use one boiler, the other not being in working order.

Condition of the boilers after three months' work. 57) In consequence of this state of affairs Mr. THORNYCROFT requested the Spanish Admiralty to allow him to have the boilers of both boats in their actual condition inspected by one of his staff. The survey was held in March 1889 and it was found that all the tubes were covered with a coating of soot 3 to 4 mm thick increasing to 120 mm in the lower parts which were only accessible after removing the boiler casing and the bunker sides. The bottom side plates of the casing were destroyed and the steam-pipes for blowing out the soot were stopped up and partially corroded. The ash-pits were filled up with soot and portions of the bridges and fire-brick lining burnt away. The zink plates in the upper drum had wasted about 3 to 4 mm from their original thickness and there was no sign of internal corrosion of the plating. In the bottom drums which also shewed no internal corrosion the zink plates were not so much attacked, but considerable quantities of deposit lay at the bottom. The tubes were tight, with the exception of one, the second from the outside in the eleventh row from the front in one of "Ariete's" boilers, a hole, 6 mm in diameter having formed near the top, which was the reason why the boiler could not be worked.

Plugging leaky tubes. 58) Mr. THORNYCROFT considered that the engineers should have cut out this tube at sea and stopped the ends in the tube-plates with the plugs provided for the purpose. According to Mr. SCOTT, the well-known shipbuilder of GREENOCK**), the plugging of defective tubes is more easily talked about than done. He has been endeavouring since the beginning of the sixties to introduce water-tube boilers into steamers and has fitted a

*) Revista general de Marina 1888.

**) Transactions of the Institution of Naval Architects. London 1889. p. 279.

considerable number, without however being satisfied with their results. In traversing Mr. THORNYCROFT'S remarks he quotes the case of one of his steamers of about 800 tons d. w. which was lying loaded at the quay ready to start when suddenly a little hole opened in a single tube of one of her water-tube boilers. The steam escaped with such violence that it was impossible to remain in the engine-room or stoke-hole until all the water in the boiler was evaporated. As it was not exactly unusual in Mr. SCOTT'S experience for tubes to pit through in this way there is evidently a considerable risk of its occurring at any time with water-tube boilers although their excellent circulation reduces internal corrosion to a minimum.

59) It being known that steel tubes are much more liable to corrosion and pitting than the former iron ones, Mr. THORNYCROFT tried replacing them with tubes of copper and of brass. Both these experiments failed on account of the low strength of the materials at high temperatures and the great liability of the upper parts of the tubes to become over-heated. Subsequently Mr. THORNYCROFT adopted internal galvanizing of the tubes, having after many trials succeeded in perfecting the process which was gone through after the tubes were bent to shape. It has however been found out since that the thin coating of zink is attacked by the acids in the boiler water, resulting if the boiler is for any considerable time off work, in an accumulation of hydrogen and consequent risk of explosion. The internal galvanizing has since been abandoned. **Internal galvanizing of the tubes.**

60) More serious than internal corrosion is the external rust and wasting brought on by the use of the steam jet for removing the soot and this Mr. THORNYCROFT proposes to prevent by galvanizing the tubes on the outside. Admitting that the Spanish engineers were very negligent, it must nevertheless be borne in mind that all the soot among the tubes cannot be blown away up the funnel by means of the steam jet. A portion of it will always fall down and, combining with the moisture of the steam used for the jet, form a deposit which adheres to the lower parts of the tubes while the boiler is laid off and corrodes them seriously, especially when the coal used contains sulphur. In "Ariete's" boilers the thin casing plates were completely destroyed by these accumulations of damp soot according to Mr. THORNYCROFT'S engineer's report. **External corrosion.**

61) External corrosion can only be avoided by thoroughly clearing all soot off the tubes and drying them completely every time the boiler is laid off. But this thorough cleaning necessitates **Proper treatment of Thornycroft boilers.**

not only the removal of the casing but also that the boiler should be accessible on all sides. A torpedo-boat with THORNYCROFT boilers must therefore be laid off for one or two days after every trip of any considerable length to thoroughly clean her tubes, over and above the day of rest necessary for the crew; and these stoppages must be made at much shorter intervals than would be required for changing the water. To make the boiler casing accessible all round it is necessary to encroach on the bunker-space which is always cramped in torpedo-boats, but without measures of this sort it is impossible to keep the boilers up to their highest pitch of efficiency.

Prospects of the Thornycroft boiler.

62) Other injurious influences affecting the life of water-tube boilers, as for instance the formation of fatty acids, are combated by limiting the use of cylinder oil, filtering the feed-water, fitting zink plates, using soda, &c., just as the burning of the tubes consequent upon internal deposit is prevented by distilling the feed-water. But all these precautions make very heavy demands upon the watchfulness and patience of the engine-room staff. If they possess these qualities or if they can attain them by thorough training, the THORNYCROFT boiler may displace the present locomotive type.

Description.

63) **The Normand*) boiler of 1890** is only the DU TEMPLE boiler in another shape. The top drum which is provided with a dome is connected with two bottom drums by a number of copper and brass tubes, part of which are "drowned" see Figs. 9 and 10, Pl. 32. At the back end of the top drum there is a large down-comer divided at the lower end into two branches which conduct the feed-water into the bottom drums. As the boiler does not differ much in other respects from THORNYCROFT'S it is unnecessary to describe it further. It has not only been fitted to French torpedo-boats by NORMAND but also by PALMERS and J. & G. THOMSON to British torpedo-boat destroyers. — NORMAND**) always works his boiler in connection with a surface-feed-heater, raising the feed to a temperature of over 115° C. and taking steam for this purpose from the L. P. steam-chest. To this intense heating of the feed as well as to the unusually high compression adopted in the cylinders of his engines (which he provides with special escape valves) NORMAND entirely attributes the economical results of his latest torpedo-boats. He states that up to 15 knots they only burn 0.5 kg and up to 25 knots only 0.9 kg of coal per *IHP* per hour.

*) The Engineer 1892. II. p. 46.

**) Transactions of the Institution of Naval Architects. London 1895. p. 34.

64) In May 1891 the French torpedo-boat No. 149, the boiler of which is illustrated here, attained 23.58 knots on three measured mile runs, her length being 36 m, displacement 76 tons, area of midship section 3.246 sq. m, revolutions 325. The engine was a triple. On a subsequent two hours' run she did 24.5 knots with 339 revolutions and 12.5 atmos. working pressure. It is reported that the boiler weighed 9.6 tons and that the maximum *IHP* was 1332, shewing about 400 *IHP* per sq. m of grate and about 140 *IHP* per ton of boiler at a combustion of 340 kg per sq. m of grate per hour. This quite unusually high performance must be considerably reduced if, instead of the maximum we take the mean *IHP* which is not stated, and if to the nett weight of the boiler proper and the water, that of all the fittings, bars, uptakes, &c. is added. On this basis the performance is probably but little better than THORNYCROFT'S or YARROW'S. — In the spring of 1895 THOMSONS*) of Clydebank delivered the British destroyers "Rocket", "Shark", and "Surly", each having four Normand boilers and engines of 4200 average *IHP* on trial. The grate surface being 15.33 sq. m, this gives 274 *IHP* per sq. m of grate. The weight of the boilers is not published so that the *IHP* per ton cannot be determined. These boilers are reported to have worked satisfactorily on their trials with the exception of "Shark's", some of the (copper) tubes of which burst and scalded several stokers. NORMAND in conjunction with SIGAUDY**) has recently improved his grouped boilers by fitting wide connecting pipes between the top drums as well as the bottom drums of the members of a group. This is said to overcome the oscillations of pressure and water-level which are continually occurring in boilers with small water-space, and interfere with the proper distribution of the feed among the boilers. When several boilers have one feed pump in common there is a tendency for all the water to go into the boiler in which the steam is lowest. Although a group of boilers connected as above described really form a single boiler, the plan has now been adopted of feeding each boiler separately in order to keep the steaming regular. **Results of and opinion on the Normand boiler.**

65) FLEMING & FERGUSON'S boiler***) like NORMAND'S bears a strong resemblance to the THORNYCROFT boiler of 1889. The only essential difference is in the curvature and arrangement of the tubes, but some of the details vary also, for instance accordingly as the boiler is to have one, two, or three furnaces, two, three, **Blechynden and Fleming and Ferguson's boilers.**

*) Engineering 1895. II. p. 630.

**) Engineering 1895. II. p. 541.

***) Transactions of the Institution of Naval Architects. London 1894. p. 298.

or four bottom drums are fitted from which the tubes ascend to the top drum common to them all. BLECHYNDEN'S boilers*) were fitted by the BARROW Co. to their destroyers and were satisfactory. Their tubes are grouped in such a manner that they can be removed through holes placed opposite their ends in the shell of the top drum and closed by screwed plugs, thus the cutting out of the neighbouring tubes to a defective one, as in THORNYCROFT'S boiler, is not necessary.

Description. 66) q. **The Yarrow****) **boiler of 1890** does not, strictly speaking, belong to this group as its tubes are not curved but straight. They connect the two bottom drums with the top drum which is bolted together in two halves. Each bottom drum forms the lower half of a horizontal cylinder, the upper half of which is cut away square to the tubes and replaced by a flat bolted cover, the bottom tubeplate. By this means the whole of the tubes are rendered accessible for inspection and cleaning. Mr. YARROW makes a special point of the accessibility of the inside of the tubes. This is however obtained at the cost of some of the elasticity of the boiler and therefore increases its sensitiveness to rapid heating and cooling.

Circulation. 67) The first of the straight-tube YARROW boilers were fitted with down-comers of large diameter as introduced by DU TEMPLE, and to them alone NORMAND ascribes the excellent circulation of all the boilers of this group. He therefore considers these down-comers one of the most important inventions in the whole history of boiler-making. YARROW found however that they are not at all necessary and has therefore omitted them in his latest boilers. The torpedo-boat boilers by the VULCAN Co. of Stettin (see 68) are also without down-comers and have nevertheless a good circulation. The celebrated Mr. MAXIM explains this as being due to the difference between the heat in the tubes next the fire and those at the outside. The latter always contain cooler water than the inner tubes where the water is hot and mixed with steam, so that the outer tubes perform the duty of down-comers of themselves. MAXIM therefore regards these down-comers, so highly esteemed by NORMAND, as utterly superfluous. The views of eminent practical engineers are not more uncompromisingly opposed on this point than upon another also relating to circulation. Mr. THORNYCROFT considers it necessary that the mouths of the tubes in the top drum should all be above the water line, as according to his experiments***) the circulation is

*) Transactions of the Institution of Naval Architects. London 1894. p. 303.

**) Engineering 1891. I. p. 79.

***) Transactions of the Institution of Naval Architects. London 1894. p. 287.

then doubly as active as when they are "drowned". His opponents — and they are all the other makers of boilers of this type — in no wise dispute this fact, but assert that in their own boilers even when most forced and without the down-comers which Mr. THORNYCROFT also considers indispensable, the circulation is perfectly satisfactory and the tubes being all drowned there is no danger of their upper parts which chiefly contain steam, becoming overheated.

68) Mr. YARROW*) further states that on one of the trials of the British torpedo-boat destroyer "Hornet" a defective tube gave out and was replaced in 40 minutes, also that as a test of elasticity one of the boilers was forced to its utmost steaming capacity at his works, the fire quickly drawn, and the whole of the casing immediately removed to cool the boiler down rapidly. The casing having been replaced, steam was got up to 12 atmos. as fast as possible. After this proof the boiler shewed no change of form or any other injury. On her forced trials "Hornet" indicated just under 4000 *IP*, her boilers weighing 42.8 tons, giving 90 *IHP* per ton. But as the weight of boilers stated is most probably exclusive of funnels, uptakes, and fittings, the figure should be considerably reduced, say to 80 *IHP* per ton at the outside. (Compare 73.) The VULCAN Co. of Stettin**) fitted YARROW boilers to their Chinese torpedo-boats in 1895 and they shewed an evaporative factor of 7.6, burning 360 kg of Welsh coal per sq. m of grate per hour with 42 mm of air-pressure in the stoke-hole and 50° C. feed temperature. When the combustion was raised to 430 kg the steam in the trap at the end of the main steam-pipe contained only 1 % of moisture. As the boilers neither primed nor gave any other trouble at the trials although the stokers were not thoroughly expert, the VULCAN Co. consider the YARROW boiler to be among the best of the present water-tube boilers both as regards simplicity, accessibility, and economy. **Opinion on the Yarrow Boiler.**

69) The SAMPSON boiler***), manufactured by MAUDSLAY is a good deal allied to the YARROW boiler. It also has straight tubes, two shallow water-chambers at the bottom and two half cylindrical steam-collectors at the top. — Another boiler of this type is MUMFORD'S†). It is chiefly intended for launches &c. and consists of two bottom drums and one top drum connected by separate nests of curved tubes fitted into flat boxes. **Sampson's, Mumford's, and Petersen's boilers.**

*) Transactions of the Institution of Naval Architects. London 1894. p. 335.

**) Engineering 1895. II. p. 674.

***) Mittheilungen aus dem Gebiete des Seewesens. Pola 1894. p. 619.

†) Ibid. 1895. p. 845.

PETERSEN and MACDONALD *) in their boiler, go still further in this direction by dividing up into groups of 9 the tubes which are of 22 mm diameter. These boilers have been manufactured by Mr. HOWALD of Kiel. Although the grouping of the tubes may be very handy for renewals and examination, it is questionable whether the complicated connections involved will remain tight under the high temperatures and the unavoidable racking.

Description. 70) r. **The White Boiler of 1893** **), Figs. 3 to 5, Pl. 33, is a combination of THORNYCROFT'S boiler of 1889 with NORMAND'S and BELLISS'S of 1888 ***). The arrangement of one top and two bottom drums is taken from THORNYCROFT, the down-comer between them at the back from NORMAND, and the spiral form of the tubes from BELLISS. The grate is placed very low, giving a good combustion-space. The front and back of the casing are protected by closely spaced tubes, the curvature of which is in one plane only, as shewn on the right hand side of Figs. 3 and 5. It will also be seen from the latter figure that each side of the boiler is provided with a row of slightly bent tubes placed between the spiral tubes and serving to carry the top drum, as the left side of Fig. 3 also indicates. These tubes as well as those at the front and back are contracted at their ends to get sufficient material between the holes in the tubeplates with such close spacing and act as screens to conduct the hot gases back in the direction of the arrows in Fig. 5 to the two funnels so as to use up the heat better. White's boiler is in use on several British torpedo-boats and no one will dispuste its elasticity but it is stated that a portion of the spiral tubes get burnt through after a comparatively short period of work.

71) REED'S†) boiler is also closely allied to THORNYCROFT'S. The tubes are curved and the inside rows are bent into a zigzag from front to back at their lower parts. Besides this, the attachment of the tubes in the top and bottom drums is worthy of remark. It is shewn in Figs. 18 and 19, Pl. 39 and consists of a nut with a hemispherical end screwed on the tube and bedding into a recess of corresponding form in the tube-plate; the tube passes through the tube plate and has a check-nut inside. These boilers were fitted by PALMERS' Co. to their destroyers and have answered well.

Description. 72) s. **The Thornycroft boiler of 1893** is distinguished from its predecessor described in 54) by having only one bottom drum instead of two (see Figs. 7 and 8, Pl. 32). In the centre between the top and bottom drum there is a row of large tubes bent S-shaped

*) The Engineer. 1895. II. p. 366.

**) Transactions of the Institution of Naval Architects. London 1894. p. 303.

***) C. BUSLEY. Die Entwickelung der Schiffsmaschine. Edit. III. Berlin 1892. p. 199.

†) The Engineer. 1896. I. p. 172.

(Fig. 7) which perform the office of the former outside downcomers. On both sides of these are grouped the water-tubes leading from the bottom drum to the steam-space of the top drum, and beyond these on each side there is a roomy combustion chamber with very considerable grate surface. The external boundary of the boiler is formed by a row of closely-spaced tubes with their top ends entering the steam-space of the top drum and their lower ends connected to a large water-tube bent to a U shape and extending on both sides from the back end of the bottom drum. The separation of the steam from the water carried over is effected by little angled plates fixed in front of the mouths of the tubes in the steam-space, Figs. 20 and 21, Pl. 39, placed so that each plate comes opposite an interval between two in the next row. The external casing of the boiler is of sheet steel covered with asbestos mill-board. The hot gases pass at the bottom through the spaces of the centre nest of tubes because their passage upwards among the external middle tubes is stopped by the close spacing. Afterwards they pass from the middle upwards and backwards into the funnel. An automatic feed regulator described in § 69 is fitted. In devising these boilers Mr. THORNYCROFT had two objects in view, first to obtain more grate surface than was possible in his earlier boilers, and secondly to produce a boiler which would readily lend itself to being massed in numbers in cases where the greatest production of power on the smallest possible weight is desired.

73) Mr. THORNYCROFT*) states that these boilers, first fitted to "Daring" and then to a number of other British destroyers, have worked well and without priming since the feed regulators were adopted. With a total weight of 48.5 tons including water but without funnels they produced 4409 *IHP* on a forced trial, or 91 *IHP* per ton. On the sister ship "Decoy" where the power was only 4049 *IHP*, the figure falls to 83 *IHP* per ton. But to compare these rates with those given for other boilers, they must both be somewhat reduced, as the funnels and all stokehole fittings are left out of account. The average performance on three-hour forced trials may therefore be put down as not exceeding 80 *IHP* per ton, thus beating the best of the locomotive boilers by 30 *IHP* per ton. But here it must be noted that all the objections to the former THORNYCROFT boiler with respect to accessibility, cleaning, and up-keep (see 58 to 61) apply equally to this later one. It is nevertheless *certain that the Thornycroft boiler is particularly well adapted for obtaining splendid trial-trip results; if it continues to answer as well*

Opinion upon the new Thornycroft boiler.

*) Proceedings of the Institution of Civil Engineers. April 1895.

in regular work as on the trials of the recent British destroyers it must be considered to mark distinct progress in the construction of marine water-tube boilers.

Description. 74) **t. The Ward boiler of 1888** *) (Figs. 6 and 7, Pl. 33) is fitted to the U. S. coast-defence ship "Monterey". To one central vertical drum serving as a steam-collector two opposite rows of vertical cast steel tubes of large diameter are fitted. On the one side of the drum these tubes rest upon a horizontal tube connected to their lower ends; their top ends are closed (Fig. 6, right). On the other side of the drum the vertical tubes are connected at both ends with a horizontal tube. The three horizontal tubes are united at their inner ends to the drum, their outer ends are closed. The vertical tubes are connected in pairs one tube on each side of the drum, by a series of semicircularly bent tubes which lie in slightly inclined planes and are attached with right and left-handed unions to short branches screwed into the vertical tubes with left-handed threads, so that each tube can be separately taken out.

Feeding and circulation. 75) The feed-pipe enters at the centre of the bottom of the middle drum and is carried up internally to about the water line where it terminates in a rose directed downwards. The water thence flows downwards and most of it enters the bottom horizontal tube on which the close-ended vertical tubes are placed, the rest of it passing into the other horizontal tube with the open-ended vertical tubes. From the close-ended vertical tubes the water flows through the semicircular tubes into the open-ended vertical tubes and passes off thence as steam through the top horizontal tube into the steam-space of the middle drum and then into the steam-pipe through a perforated dash-plate which throws down the water carried over. As all the parts of the boiler can expand separately, there is sufficient elasticity. On removing the casing top and the top horizontal tube, each pair of vertical tubes can be taken out for examination and repairs. The grates are of cast iron in a number of sections of which "Monterey" has 24.

Trial-trip results. 76) The iron-clad "Monterey" has two single-ended Scotch boilers and four WARD boilers distributed in two stoke-holes divided by a longitudinal bulkhead and each having two WARD and one Scotch boiler. The four WARD boilers were intended to give 4200 *IHP*, their power was even estimated at 4500 *IHP***) from some evaporative trials made by the well-known Engineer LORING. On the four-hours' forced trials***) however, only 5244 *IHP* could be obtained on an average, including auxiliary

*) Journal of the American Society of Naval Engineers. Washington. 1893. p. 127.

**) Ibid. Washington. 1890. p. 416.

***) Ibid. Washington. 1893. p. 115.

machinery, from all the boilers together, 1200 of which were put down to the Scotch boilers, — whereas 5400 *IHP* were anticipated. As the latter had 8.17 sq. m of grate surface and got the 1200 *HP* expected of them at the rate of 150 *HP* per sq. m, it can only be assumed that the WARD boilers did not get up to the required 4200 *IHP*; for they would have produced 4100 at 150 *HP* per sq. m with their 27.4 sq. m of grate. But even assuming that they did reach 4200, it would only have amounted to 60 *HP* per ton of their 69.97 tons of weight, instead of 79 *HP* per ton, as given by the evaporative trials. These trials shewed an evaporative factor of 6.6, at 269 kg of coal burnt per sq. m of grate per hour, 10^0 C. temperature of feed, 100^0 temperature of steam, and 12.48 $^0/_0$ moisture of steam, while the torpedo-boot "Cushing's" THORNYCROFT boiler tried at the same time reached an evaporative factor of 7.53 at a combustion of 196 kg per sq. m and 3.89 $^0/_0$ of moisture of steam. The water in the WARD boiler used at these trials weighed 2.01 tons, or in round numbers $^1/_7$ of the total weight, 13.85 tons. It is reported that there was difficulty in feeding the WARD boilers regularly on "Monterey's" forced trials. At any rate the U. S. naval authorities rested on their oars for some time after this first experiment with water-tube boilers; it has only quite lately been decided to fit the "Chicago", a comparatively old cruiser with BABCOCK-WILCOX boilers, while for the latest torpedo-boats the MOSHER water-tube boiler (see 36) has been adopted.

77) **IV. Efficiency of Water-tube Boilers.** As previously shewn, there is an essential difference between straight-tubed and curved-tubed water-tube boilers, for their capacity and efficiency depend upon the kind and arrangement of the tubes. The efficiency of water-tube boilers, like that of locomotive boilers is to be calculated for the hardest forcing, first to enable a comparison to be made between these two types, and in the next place because water-tube boilers have hitherto been chiefly used in war-ships in the place of locomotive boilers. Efficiency.

78) a. **The actual efficiency** of straight-tubed is considerably higher than that of curved-tubed boilers, because the former cannot be forced so hard as the latter. The present straight-tubed boilers mostly drive triples at 11 to 14 atmos. absolute initial pressure. Not much more than 150 kg of coal can be burnt per sq. m of grate per hour, producing about 140 *IHP*. The steam consumption with the high cut-offs used is seldom less than 8 kg per horse per hour. Therefore 1 kg of coal evaporates Actual efficiency of straight-tubed water-tube boilers.

$$\frac{140 \times 8}{150} = 7.45 \text{ kg of water,}$$

and if 1 kg of coal can theoretically evaporate 14 kg of water at these high pressures, we get an actual efficiency of

$$\frac{7.45}{14} = 0.53.$$

Actual efficiency of curved-tubed boilers.

79) Curved-tubed boilers work either triples or quadruples of 15 to 18 atmos. absolute initial pressure. They burn up to 350 kg of coal and produce 240 *IHP* per sq. m of grate per hour. The steam consumption of the engines must again be put at 8 kg especially as the tubes of these boilers generally have to be swept with a steam jet. Accordingly 1 kg of coal must evaporate

$$\frac{240 \times 8}{350} = 5.5 \text{ kg of water.}$$

At a theoretical evaporation of 14 kg, the actual efficiency is therefore

$$\frac{5.5}{14} = 0.40.$$

Efficiency according to Rankine.

80) b. **The efficiency according to Rankine** of the straight-tubed boilers with their average ratio of heating surface to grate of about 30:1 comes out

$$\frac{Q_1}{Q_0} = 0.92 \frac{30}{30 + 0.1 \times 150} = 0.61.$$

Curved-tubed boilers have mostly a ratio of heating surface to grate of 45:1, so that we get

$$\frac{Q_1}{Q_0} = 0.92 \frac{45}{45 + 0.1 \times 350} = 0.52.$$

Efficiency according to Wilson.

81) c. **The efficiency according to Wilson** of the straight-tubed boilers is similarly obtained by substituting the value of the coefficient at 0.8 instead of 0.92,

$$\frac{Q_1}{Q_0} = 0.8 \frac{30}{30 + 0.1 \times 150} = 0.53,$$

and for curved-tubed boilers

$$\frac{Q_1}{Q_0} = 0.8 \frac{45}{45 + 0.1 \times 350} = 0.45.$$

The efficiency of the highly-forced curved-tubed boiler is just as low as that of the locomotive type. If however water-tube boilers are worked moderately their efficiency rises and they are nearly as economical as Scotch boilers.

Table.

82) The table on p. 576 contains the principal numerical values for forming an opinion upon the performance of a marine boiler. Those in the first five columns are based upon the results of natural-draught trials, those of the last four refer to conditions of the hardest forcing as the boilers are expressly intended to work under them.

83) **V. Advantages and disadvantages of water-tube boilers.** The following advantages are in general claimed for water-tube boilers: Advantages and disadvantages.

a) light weight,
b) small space occupied,
c) small first cost,
d) small cost of repairs,
e) slight risk of explosion,
f) high economy.

To these good qualities two others are added which are of special value for naval purposes,

g) rapidity of getting up steam,
h) capability of bearing repeated hard forcing.

Their disadvantages may be stated as

i) tendency to prime,
k) difficulty in feeding,
l) sensitiveness to corrosion and dirt,
m) impossibility of plugging defective tubes under steam.

These several points will now be considered under the light of experience.

84) a. **Light weight.** Water-tube boilers are lighter than any others which is chiefly due to their small quantity of water. On the other hand the masonry about their furnaces is many times heavier than in other boilers, while their other fittings, uptakes, and funnels, as well as stoke-hole gear and firing tools, all of which must be included in the weight of the boiler itself to make up the *working weight* of a water-tube boiler, are no lighter than those of other types of boilers. Taken in proportion to their power, water-tube boilers are no lighter than others. The straight-tubed French boilers, as BELLEVILLE'S, D'ALLEST'S and NICLAUSSE'S, have as yet produced not quite 30 *IHP* per ton of working weight on forced draught trials, while 25 *IHP* per ton and above have been got out of Scotch boilers on British war-ships. From locomotive boilers on the other hand over 50 *IHP* per ton have been obtained, for "Satellit" built by SCHICHAU shewed 51 *IHP* per ton. Only curved-tubed boilers like THORNYCROFT'S, NORMAND'S, &c. have reached about 80 *IHP* per ton. Light weight.

85) b. **Small space.** In most ships it is more difficult to obtain the floor-area necessary for placing the boilers and rendering them generally accessible than it is to get the overhead room they require, and for this reason the floor-area and not the cubic space of the various boilers per *IHP* is here placed in comparison. As the engines are in every case triples of 12 atmos. working pressure, the following table indicates that the space saved by BELLEVILLE'S Small space occupied.

Table of the most important ratios relating to marine boilers.

Ratio	Box boiler (low-pressure engines)	Early Navy Boilers (compounds)	Later Navy Boilers (triples)	Early Scotch Boilers (compounds)	Later Scotch Boilers (triples)	Early locomotive Boilers (compounds)	Later locomotive Boilers (triples)	Straight-tubed Water-tube Boilers (triples)	Curved-tubed Water-tube Boilers (triples & quadruples)	Remarks
1	2	3	4	5	6	7	8	9	10	11
Working pressure in atmos.	2	4—7	10—12	4—7	10—13	10—12	13—15	12—15	14—20	The values in the columns are averages, 2 to 6 refer to natural, 7 to 10 to the severest artificial draught.
IHP per ton of weight of Boiler under natural draught	10	13	18	12	16	26	30	18	40	
IHP per ton of weight of Boiler under artificial draught	12	19	30	18	25	37	50	30	80	
IHP per sq. m of grate on forced trial	100	110	130	85	140	220	350	140	240	
Feed-water per IHP per hour in kg.	13	10	7.5	10	6.5	8 0	8.0	8.0	8,0	
Water evaporated per sq. m of grate per hour in kg.	1300	1100	975	850	910	1760	2800	1120	1920	
Ratio of heating surface to grate	30	25	32	25	33	40	60	30	45	
Greatest weight of coal burnt per sq. m of grate per hour in kg.	150	135	120	105	110	300	500	150	350	
Water evaporated per kg. of coal per hour in kg.	8.7	8.2	8.1	8.1	8.3	5.9	5.6	7.5	5.5	
Actual efficiency	0.60	0.57	0.58	0.56	0.60	0.42	0.40	0.53	0.40	
Rankine's efficiency	0.61	0.60	0.67	0.65	0.69	0.52	0.46	0.61	0.52	
Wilson's efficiency	—	0.52	0.58	0.56	0.60	0.45	0.43	0.53	0.45	

as compared with Scotch and locomotive boilers is not worth notice and the other type of THORNYCROFT'S does not even come equal to the locomotive. The recent THORNYCROFT boilers*) on the other hand, when grouped afford a saving of floor-area as compared with locomotive boilers. In France it appears to have been found out that there is no space saved with water-tube boilers, as the new first-class cruisers of 8500 to 9000 tons displacement are to have double-ended Scotch boilers**) *if want of space renders it impossible to fit water-tube boilers*, which happened in the first-class cruiser "D'Entrecasteaux".

Table of Floor-area occupied by various Boilers.

Description of Ship	Name of Ship	*IHP*	Mean breadth of boiler-room m	Total length of boiler-room m	Total area of boiler-room-floor	*IHP* per sq. m of boiler-room floor	Number and description of Boilers
1	2	3	4	5	6	7	8
		I. Scotch Boilers.					
German Cruiser	"Kaiserin Augusta"	14015	14.6	33.4	488.4	28.7	8 Double-ended by the Germania Co.
German Iron-clad	"Wörth"	10228	12.3	28.5	350.5	29.2	12 Single-ended by the Germania Co.
		II. Locomotive Boilers.					
German Iron-clad	"Beowulf"	4866	7.87	20.25	159.4	30.5	4 Boilers by the Weser Co.
Austrian Torpedo-cruiser	"Satellit"	4792	4.80	17.50	84.0	57.0	4 Boilers by Schichau
		III. Water-tube Boilers.					
French Dispatch-boat	"Alger"	5089	8.10	30.15	244.2	20.8	24 Belleville*)
" Cruiser	"Latouche-Tréville"	8300	8.20	25.28	207.3	40.0	16 "
" Dispatch-boat	"Léger"	2170	4.42	15.70	69.4	31.3	6 "
" Iron-clad	"Brennus"	13810	14.94	21.20	316.7	43.6	32 "
British Torpedo-cruiser	"Speedy"	4703	5.72	20.80	119.0	39.5	8 Thornycroft**) (Early Type).

*) Journal of the American society of naval engineers. 1890. p. 566.
**) Transactions of the Institution of Naval Architects. London 1894. p. 331 & Plate XLVIII.

86) c. **Small first cost.** A water-tube boiler is lighter, ex fittings &c., Small first cost.
than a cylindrical boiler as it contains less material (mostly steel) and is therefore likely to cost less. Besides, water-tube boilers consist of a large number of repeat parts which can be cheaply manufactured by specially-adapted machinery, and the cost of erection of these parts is only slight compared with the boiler-makers' labour on cylindrical boilers, so that it is remarkable that a set of water-tube boilers is not only relatively but absolutely more costly than one of Scotch or locomotive boilers. The following table shews a comparison between several

*) Transactions of the Institution of Naval Architects. London 1894. Plate L.
**) Mittheilungen aus dem Gebiete des Seewesens. Pola 1895. p. 863.

sets of boilers of equal power, the particulars of the French boilers being taken from the journal quoted below*), while those of the double-ended and locomotive types are the contract prices at the respective German yards and THORNYCROFT'S that of an English contract. According to this statement, water-tube boilers are about twice as dear as Scotch per ton of weight, but not quite in that proportion per horse-power. Even if the figures may come a little more favourable to water-tube boilers on taking a wider range than that of the table for comparison, so much at any rate is certain that no considerable reduction in first cost is to be hoped for from them.

Cost of Water-tube Boilers.

Name of Ship	Number and Description of Boilers	Working pressure in atmos.	Admission pressure in H. P. Cylinder in atmos.	Grate-surface in sq. m	Heating surface in sq. m	Weight of Boilers in tons	Cost of the Boilers	Cost per ton	Cost per IHP
Cost of a Set of Boilers for 9000 IHP.									
							£	£	£
"Bugeaud"	24 Belleville	17.0	12.0	70.15	2000	273.3	24142	88.3	2.68
"Chasseloup-Laubat"	20 D'Allest	15.0	12.0	68.00	1807	229.2	20132	87.8	2.23
"Friant"	20 Niclausse	15.0	12.0	72.72	2168	227.4	24606	108.2	2.73
—	4 Scotch Double-ended	12.0	12.0	66.00	2000	358.0	18800	52.5	2.09
Cost of a Set of Boilers for 4800 IHP.									
—	8 Thornycroft	12.0	12.0	25.25	1500	135.0	14750	109	3.07
—	4 Locomotive	12.0	12.0	20.35	1100	138.5	9950	76.8	2.07

Small cost of Repairs.

87) d. **Small cost of repairs.** As a rule the only repairs required in a water-tube boiler are replacing defective tubes and therefore cost but little in any particular case. The experience in the French Navy**) is however that such repairs are much more frequently wanted than repairs of any kind are in cylindrical boilers and that the upkeep of water-tube boilers is really the more expensive. VOGT***) corroborates this in his Report for 1890 and 91 upon the stationary boilers under his inspection, stating that 10 % of the water-tube boilers had required considerable repairs and only 2.3 % of the ordinary boilers. The higher cost of repairs is to a certain extent made up for by the short time in which they can be carried out. A BELLEVILLE element can be replaced at sea in two hours including blowing the boiler out and getting up steam again and a tube can be still more

*) Journal of the American Society of Naval Architects. Washington 1895. p. 408.

**) Ibid. p. 367.

***) Geschäftsbericht des Bergischen Dampfkesselrevisionsvereins zu Barmen 1890 and 1891.

expeditiously exchanged in a NICLAUSSE or DÜRR boiler. In D'ALLEST'S this operation is more difficult as the tubes are expanded in at both ends, and it is still worse in the curved-tubed boilers. In THORNYCROFT'S for instance five hours are necessary. Besides the facility of repairs due to the boilers being put together of many small pieces there is the great advantage that in case of renewal of a whole boiler, its component parts can easily be passed down through the boiler hatch and connected up on board. The tearing up of decks &c. necessary for renewing the boilers in most ships, is thus dispensed with, saving much time and money, especially in war-ships with armoured decks.

88) e. **Slight risk of explosion.** Although serious explosions which might endanger the vessel's hull, are hardly to be expected and nothing happens as a rule but the bursting of one or more tubes or the blowing out of a joint, still these occurrences are serious enough for the engine-room staff. This is evident from the experience gained with stationary water-tube boilers. Two superintendent engineers who were responsible for 4450 boilers stated at the Congress of the International Union of Boiler Inspection Associations at Danzig in 1891 *) that explosions of water-tube boilers cause as a rule little destruction of property but that such boilers are subject to a relatively great number of casualties often seriously injuring the attendants. This report is also strengthened by the statistics of boiler explosions of the German Empire during the years 1892 to 94. A BABCOCK-WILCOX boiler with which some trials were being made by the FAIRFIELD Co. in 1895 had several tube-end doors blown off whereby six persons were injured although they were able to get out of the trial shed at once. On the trial-trip of the British torpedo-boat destroyer "Shark" built by J. & G. THOMSON in 1894, several tubes burst and some of the men were scalded. In the face of these facts it cannot be asserted that "water-tube boilers enjoy complete immunity from explosion". **Slight risk of explosion.**

89) f. **High Economy.** All water-tube boilers contain a small quantity of water in spaces of small section with very thin walls, almost the entire surface of which is in contact with the hot gases. On reducing the diameter and increasing the number of the tubes they can be made thinner, the heating surface augmented, and consequently the heat more completely used. As besides this, the superior water-tube boilers have for the most part a good circulation, they ought to be particularly efficient evaporators. But although the average consumption **High economy.**

*) Zeitschrift des Vereins deutscher Ingenieure. Berlin 1892. p. 1226.

with Scotch boilers is only 0.75 kg of coal per *IHP* per hour, the French straight-tubed water-tube boilers have not yet, according to the trial-trip reports, got down as low as 0.8 kg, even at reduced power. The lowest consumption yet attained is that of the THORNYCROFT boilers of the British torpedo-boat destroyer "Ardent", viz. 0.694 kg per *IHP* per hour when indicating 500 *IP* and steaming 13 knots, as against 4306 *IHP* and 28 knots on the forced trials. But when eased down to this extreme extent, cylindrical boilers have shewn still lower consumptions. — This low economy of water-tube boilers is due to the imperfect combustion which takes place in most of them. Many have only *one* large grate on which the irregularities in the air-supply have a much worse effect than if the grate were divided among several furnaces capable of being fired independently of each other. In the closed furnaces of Scotch and locomotive boilers the bridge affords a means of detaining the gases sufficiently long to get properly mixed with air and completely burnt. In most water-tube boilers the gases escape among the tubes immediately they are generated and having no opportunity afterwards to mix with the necessary air are therefore only partially consumed. In BELLEVILLE boilers, for instance, the combustion is sometimes so bad that it has been found necessary to introduce jets of compressed air at about $^1/_3$ atmos. pressure into the upper part of the furnace, which on the one hand conveys to the rising gases the oxygen they want, and on the other hand detains them on the grate until they are consumed; the air for the purpose is compressed in a special pump. D'ALLEST'S furnaces are the best arranged in this respect which accounts for the superior economy of his boiler over BELLEVILLE'S and NICLAUSSE'S. — There is as yet no experience to shew that Scotch boilers are less economical than water-tube boilers.

Rapidity of getting up steam.

90) g. **The rapidity** with which steam can be got up is incontestably very great in the curved-tubed boilers. One of "Daring's" THORNYCROFT boilers was heated in the author's presence so rapidly with a wood fire that steam was got up to 6.66 atmos. in 15 minutes from lighting up. At this pressure the engines can be started, provided they are properly warmed through in time. For torpedo-vessels, fast cruisers, and blockade-runners this quality of curved-tubed boilers has an indisputable value. In straight-tubed boilers steam cannot be raised so quickly, for BELLEVILLE'S require three quarters of an hour and then become leaky if the process is often repeated, so that the rule in the Messageries Maritimes fleet is to allow 2 hours, which time is

only to be shortened to $1^1/_2$ hours is case of urgent necessity. The conditions are better in the more elastic boilers of DÜRR and NICLAUSSE, but much worse in D'ALLEST'S. The latter can get up steam in about an hour but then leaks in the tube-ends and joints begin to shew and after repeated rapid heating up get so bad that the fires have to be drawn. The usual time for getting steam on this boiler is four hours, but six are better which will almost suffice for Scotch boilers, as six to eight hours are now generally allowed for them.

91) h. **The capacity to bear repeated hard forcing** is only possessed by the curved-tubed boilers, for the straight-tubed ones require as considerate treatment as Scotch and locomotive boilers, neither of which can withstand for any length of time either frequent hard forcing or sudden variations in the call for steam. Tube-ends leak, combustion-chamber stays are broken, and finally the seams are started in the most exposed places. It has therefore become the rule not to exceed 30 mm air-pressure for Scotch and 50 mm for locomotive boilers. The corresponding weight of coal burnt per sq. m of grate per hour on a six-hours' trial-trip is not above 150 kg for Scotch and barely 250 kg for locomotive boilers, whereas twice that quantity was formerly burnt in the latter. In the French straight-tubed boilers 150 kg per sq. m of grate have not yet been reached on four-hours' forced-draught trials although in DÜRR boilers 200 kg have been got. In "Bruiser's" THORNYCROFT boilers 330 kg were burnt during a three hours' forced run, in NORMAND'S 350, and in YARROW'S (see 68) 430, also lower than in the locomotive boiler. — While YARROW made his elasticity experiment described in 68, THORNYCROFT *) emulated it by blowing out two of "Daring's", boilers, afterwards partially drawing the fires, and then pumping up the boilers with cold water. They went on working without any leakage. Mc FARLAND reports some still more drastic proceedings with a TOWNE boiler in America **); the fire when being most heavily forced, was suddenly drawn and the boiler immediately played on with a hose and cold water without shewing any leak. These tests appear to prove that many water-tube boilers are less sensitive to sudden changes of temperature than cylindrical ones. Capacity for withstanding forcing.

92) i. **A tendency to prime** is common to all boilers with small water-spaces and confined water-line area. If the call for steam is variable, as when manœuvring with a fleet, and unless the feed keeps pace with the evaporation and the firing is very Priming.

*) Transactions of the Institution of Naval Architects. London 1894. p. 336.

**) Journal of the American Society of Naval Engineers. Washington. 1894. p. 664.

carefully attended to, such boilers, even if not always priming, will make wet steam. On Admiralty forced-draught trials of Scotch boilers from 500 to 600 kg of steam at 12 atmos. pressure must be developed per sq.m of water-line area per hour and in locomotive boilers this figure reaches 700 to 800, or with extreme forcing about 1000. But in BELLEVILLE boilers under equal conditions of forcing as much as 4000 to 5000 kg of steam at 17 atmos pressure per sq.m of tube orifice must pass per hour into the steam-space and in THORNYCROFT boilers this weight of steam is nearly doubled. What kind of a performance this is may be grasped by comparing it with a statement of CARIO'S *) to the effect that for steady steaming without priming in stationary boilers (not water-tube) only 100 kg per sq.m of water-line area must be allowed per hour. Water-tube boilers therefore all require some special steam-drying apparatus either of the nature of a superheater or a mechanical appliance to prevent the wetness of the steam exceeding a safe limit.

Feeding.

93) k. **The difficulty of feeding** is a consequence of the small water-space and the oscillations in the pressure it entails. BELLEVILLE, YARROW, and THORNYCROFT therefore fit their boilers with an automatic feeding arrangement actuated by a float. This sytem however has many opponents among superintendents as it gets the engineers into careless habits, increasing the risk of mishap in the event of the apparatus sticking in consequence of injury or dirt, as has been several times reported.

Corrosion and Dirt.

94) l. **Sensitiveness to corrosion and dirt** is of course consequent upon the small thickness of the tubes which are considerably weakened even by slight incipient pitting. The drums and headers of the various boilers can be protected by zink plates, but not the tubes. Copper and brass tubes have therefore been tried but had to be abandonned on account of their low strength at high temperatures and replaced with steel ones (compare 59), as for instance on "Shark" after the explosion (see 88). As an internal examination of the curved tubes is impossible under any circumstances, it has become the rule in the American Navy to renew all the tubes every three years. Besides this there is the difficulty of externally clearing the tubes of soot and ash, mostly done by means of a steam jet. It is not always possible to get sufficient supplementary feed to make up for the steam thus sacrificed and some portion of the tarry soot, peculiar to many kinds of coal, cannot be completely dislodged by the jet. In any case the whole of the soot is not got rid of by blowing it out of

*) Zeitschrift des Vereins deutscher Ingenieure. Berlin 1889. p. 203.

the funnel, a process which does not exactly contribute to cleanliness on deck, but some always falls to the lower parts of the boilers, where it gradually adheres and forms with moisture a corrosive deposit which attacks the tubes, drums, casing, &c. In the French Navy the tubes are therefore cleaned with brushes, four boilers being laid off at a time out of the twenty or more on board. This is done in regular rotation so that every boiler is cleaned after three days' work. Difficult as the operation is with straight-tubed boilers, it is worse for curved-tubed ones and in THORNYCROFT'S much soot is reported to settle in the sharp bends even after 20 to 24 hours' steaming and to be awkward to remove.

95) *The impracticability of stopping defective tubes under steam* is the most serious drawback the water-tube boiler has. A leaky tube necessitates laying off the boiler for a period of about 2 hours for DÜRR, NICLAUSSE, and BELLEVILLE boilers and this is extended to 5 hours for THORNYCROFT'S (see 87). In all curved-tubed boilers like the latter, the replacing of a tube is especially tedious because all the tubes in a transverse row which are outside a defective one must be cut out to get at it, and the tube-holes in the top and bottom drums provisionally stopped with steel plugs until an opportunity of replacing these tubes occurs. **Impracticability of stopping defective tubes.**

96) This circumstance, besides the awkwardness of cleaning, enforces the subdivision of water-tube boilers into many small units, in order that the loss of one boiler may be as little as possible felt. But the further this subdivision is carried, the smaller each separate water-space becomes, the greater is the difficulty of feeding, and the more room in the ship the boilers occupy. Although it is not to be doubted that engineering skill will by degrees overcome the constructive difficulties peculiar to water-tube boilers, it is certain that the inspection, cleaning, and general attendance will be much more tedious, inconvenient, and exhausting for the staff than with cylindrical boilers. **Concluding remarks.**

§ 58.

Auxiliary Boilers.

1) **I. Object and classification.** The auxiliary boilers of steamers have very different work to do according to the service for which the ships are intended. On the older ironclads they drive a number of auxiliary engines such as windlasses, turret rotating gear, ventilating fans, &c. as well as bilge, wash-deck, and fire-pumps. They also have to work the dynamos in port and supply steam for the steam-heaters in winter. In large passenger steamers they furnish steam for the distillers and the winches, **Object.**

on cruisers they are also often used for distilling, and on small cargo steamers for the winches only.

Position.

2) Their position varies with their work. On men of war and the larger merchant steamers they are usually placed in the stoke-hole or as near to it as possible in a recess in the bunker. They are often purposely placed higher than the main boilers, so that steam can be kept on them after the main fires are put out by water which may have got into the ship. On many of the vessels referred to they are for the above reason placed in the tween-decks and can be worked after the stoke-hole is completely flooded. Many large passenger steamers have two auxiliary boilers, a large one in the stoke-hole for the auxiliary machinery and a smaller one in the tween-decks for the steam-ovens, the steam-heaters, and the distiller. On small coasting steamers the auxiliary or donkey boiler is sometimes placed on deck.

Disuse of auxiliary boilers.

3) On board large modern vessels, both naval and mercantile, the auxiliary boiler proper has disappeared. The number of auxiliary engines of all kinds has increased in these vessels to such an extent that auxiliary boilers of the dimensions formerly customary are no longer equal to the work. In merchant steamers for instance it has become necessary for the harbour work of keeping all the winches, dynamos, and steam-heaters going, to provide a much more powerful auxiliary boiler than formerly, in fact nearly equal to a main boiler in grate and heating surface. Those responsible have therefore made up their minds, where single-enders are fitted, to use one of them in rotation for harbour service. Where double-enders are fitted either one or two single-enders are put in (usually back to back) to work both at sea and in port as occasion requires.

Interest of the plates.

4) Although auxiliary boilers are no longer of the same importance as formerly, there are nevertheless so many of them at work in comparatively old ships that the subject could not well be omitted here. In Plates 34 to 36 a number of the most notable types of these boilers are therefore shewn with the date of construction of the *first* of each description so far as this could be ascertained.

Classification.

5) Auxiliary boilers may be grouped as
   Horizontal cylindrical multitubular boilers,
   Vertical cylindrical cross-tube boilers, and
   Vertical cylindrical multitubular boilers.

Scotch auxiliary boilers.

6) **II. Horizontal auxiliary boilers.** These are generally designed as single-ended Scotch boilers with one or two furnaces (Pl. 34, Figs. 1 to 3 and 7 to 9). They have the same working pressure as the main boilers. When this equals or exceeds 10 atmos.

it is usual to fit reducing valves in the steam pipes connected with auxiliary engines intended for a lower working pressure. The small boiler mentioned in 2) and used for steam cooking only is on the other hand usually worked at 3 to 4 atmos.

7) The design of auxiliary boiler shewn in Figs. 4 to 6, Pl. 34 is remarkable, as the usual combustion chamber is wanting. It is replaced by a half-round box lined out with fire-brick. This arrangement has been largely adopted on account of the ease with which the box can be removed and the tubes examined or cleaned. The boiler illustrated has PAUCKSCH tubes which are fitted with conical joints for convenience of removal when internal repairs are required. Boiler with external combustion chamber.

8) **III. Vertical boilers with cross-tubes** were those originally fitted, the multitubular plan being adopted later. Of the many varieties of this boiler only a few can be referred to here which are the best known and have found most favour in practice, viz. Classification.

a) COLTMAN'S,
b) COLOMBIER'S,
c) FLETCHER'S,
d) MARSTON'S,
e) FIELD'S.

9) a. **Coltman's boiler,** Figs. 1 and 2, Pl. 35 is of very simple design. It has an internal fire-box, one uptake tube and two internal cross-tubes of large diameter. Mud-holes are cut in the shell opposite both ends of the cross-tubes to facilitate cleaning and there is a man-hole in the steam-space. These boilers are cheap and have therefore been largely used. WOOD'S boiler*) is very similar and has four cross-tubes, each connected to the crown by two vertical tubes enabling the steam generated in them to escape direct. Coltman's.

10) b. **Colombier's boiler,** Figs. 3 and 4, Pl. 35 is a good deal used on French cargo boats. In it the wide straight cross-tubes are replaced by several superimposed rows of small curved tubes which cross each other. The smoke-box is formed of a thin casing of sheet, through which short tubes are led, on each side alternately, to the inside of the horizontal tubes. This arrangement is said to improve the circulation. To clean the tubes the boiler shell can be unbolted at a circumferential joint above the level of the fire-door and hoisted up round the funnel. The steam-space of the boiler has therefore a cast iron cover, jointed to the up-take. This boiler has done very well at low pressures, with high pressures and frequent forcing the tubes are said to be liable to leak. Colombier's

*) The Engineer 1888. II. p. 376.

Fletcher's.

11) c. **Fletcher's boiler,** Figs. 5 and 6, Pl. 35 has a fire-box in the shape of two conical frusta with their small ends joined. The upper cone contains 4 large tubes running from the outside upwards to the centre of the fire-box crown. Two conical uptakes pass up through the steam-space. A man-hole is placed in the shell opposite the narrowest part of fire-box, also one in the steam space between the two uptakes. A similar boiler is that by BUCKLAND of Newcastle*) in which the fire-box which is capacious at the bottom is reduced upwards to a narrow dome from which the gases return through 3 flues to the fire-box (?) and then pass off through the uptake at the side. Encircling the upper part of the inside of the fire-box dome is a trough-like chamber communicating by 3 tubes with the annular water-space.

Marston's.

12) d. **Marston's boiler,** Figs. 7 and 8, Pl. 35 contains a nest of vertical water-tubes passing through an upper smoke-box. The gases rising from the hemispherical furnace travel through a conical flue and enter the upper smoke-box at the lower part of one side, pass among the tubes, and escape into the funnel on the opposite side of the boiler. It appears that the draught is sometimes too weak to carry the gases through this somewhat devious course, at any rate the steam-jet in the funnel suggests the use of artificial draught. In order to facilitate cleaning, the boiler shell is interrupted in way of the junction of the flue with the smoke-box and the opening closed with a loose plate lined with firebricks.

Renewal of donkey boilers.

13) All the above-named boilers are stocked in various sizes by the makers. They are usually to be had ranging at certain intervals from 5 to 100 *IHP* for the convenience of ship-owners who are thus enabled to replace their used-up donkey boilers without delay. These boilers in ordinary cargo steamers, often placed on deck, driven hard, and generally taken little care of. wear out pretty rapidly and as those of the stock sizes are almost as cheap as thorough repairs it is quite usual to renew them on the first signs of deterioration in order to save time.

Field's.

14) e. **The Field boiler** has been used on many German cruisers for distilling, also for the fire-engine as steam can be rapidly got up. It is for this reason still applied to many steam fire-engines ashore. A number of close-ended tubes project downwards from the firebox crown, each containing an internal suspended tube, Figs. 9 and 10, Pl. 35. The thin layer of water between the skins of the two tubes is of course rapidly evaporated and as this takes place the cooler water inside the suspended tube

*) Engineering 1888. II. p. 365.

descends and replaces that converted into steam. Rapid circulation is thus set up, rendering the boiler a quick steamer. A cast-iron baffle hung beneath the bottom of the uptake obliges the gases to disperse among the tubes on their way up to the funnel. The tubes are secured in the fire-box crown with conical joints assisted by the steam pressure. When the boiler is to be cleaned the tubes can be driven up into the steam space and taken out through the man-hole.

15) **IV. Vertical multitubular auxiliary boilers** came into comparatively general use in the middle of the eighties when the working pressure of the main boilers of the latest compounds had risen to 6 or 7 atmospheres and that of the first of the triples to 8 or 10 atmospheres. In order to keep the capacity of the auxiliary boilers in line with that of the main boilers the heating surface of the former had to be made proportionately larger than in cross-tube boilers. The multitubular system was therefore adopted, which has the further advantage of affording much greater facility for the removal of soot &c. than the other. Multitubular auxiliary boilers all resemble each other pretty closely, the arrangement of the smoke-boxes, tubes, and crown stays always remains the same, the only difference being in the furnace. The more important examples of this type of boiler are Subdivision.

a) COCHRAN'S,
b) The Flensburg Shipbuilding Co's.,
c) CLARKE CHAPMAN & Co's.,
d) The FAIRFIELD Co's.

16) a. **Cochran's boiler**, Figs. 1 and 2, Pl. 35 is very like MARSTON'S, Figs. 7 and 8, Pl. 35, in the arrangement of the furnace, the course of the gases, and the fire-trick lining of a portion of the shell. Having reached the back smoke-box the gases pass through the tubes and escape into the funnel which is placed at the side of the boiler above the fire-door. The tubes can be easily cleaned through the front smoke-box doors after taking down the back door opposite. The inside of the boiler is accessible through a man-hole in the crown. DAVEY PAXMAN .& Co. of Colchester manufacture similar boilers with rather more roomy furnaces, these being made cylindrical with dished crown instead of hemispherical which improves the combustion. — Another boiler allied to these is TINKER SHENTON & Co's. "Duplex" of Hyde. The furnace tapers somewhat and has the crown deeply dished, so that in this respect it is about half way between the two last-mentioned boilers. The flue starts from the middle of the fire-box crown and leads into a horizontal smoke-box above which the tubes are arranged. SHARPE and Cochran's.

PALMER'S boiler manufactured by ABBOTT & Co. of Newark is still more complex. This boiler forms a combination of the tubul*ar* and tubul*ous* systems, its lofty furnace being traversed by both fire and water tubes. — BARLOW'S boiler (of Rochdale) is, on the other hand, very simple, its fire-box being divided into two parts connected by a number of rather short vertical tubes through which the gases pass from the lower into the upper box and thence laterally out into the funnel.

The Flensburg.

17) b. **The Flensburg Ship-building Company's boiler,** Figs. 3 and 4, Pl. 36 has a conical furnace growing into a half round vertical smoke-box. In order to increase the heating surface somewhat, the tubes are made so long that the front tube-plate is outside the surface of the boiler shell, to which the smoke-box and funnel are attached, the latter being at the side. The bottom of the furnace is protected with fire-brick lumps laid on the bars. These have proved good steaming boilers, the combustion is satisfactory and they are easily cleaned.

Clarke, Chapman & Co's.

18) c. **Clarke, Chapman & Co's. boiler,** Figs. 5 and 6, Pl. 36 does not differ essentially from the last-described. The furnace is rather more capacious and the tubes are set at a greater rising angle. The lower edge of the back tube-plate is a tangent to the boiler shell and its upper edge is therefore within the shell. Two cross stays are introduced between the tubes to restore the loss of strength in the shell due to the interruption of it by the back tube plate. This boiler has proved very efficient.

The Fairfield Co's.

19) d. **The Fairfield Co's. boiler,** Figs. 7 to 10, Pl. 36 is nothing but a Scotch boiler enclosed in a vertical cylinder. It contains much more grate and heating surface than the boilers described above and is therefore much more powerful. As it is bounded above and below by spherical surfaces it is of very strong form and thus requires very little staying. It has only 2 cross-stays, 2 short stays to support the bottom front plate, 4 girder-stays to the combustion chamber crown, and the stays for the combustion-chamber back. The two large man-holes make the boiler easily accesible inside both at top and bottom. These boilers have always given satisfaction on the large transatlantic mail-steamers for which they have been chiefly made by the FAIRFIELD Co.

20) **V. Rules for auxiliary boilers.** In Germany the police regulations as to boilers in general cover all auxiliary boilers on board ship. In England only such auxiliary boilers on passenger steamers as are subservient to or connected with the main boilers and engines come under the Rules and Regulations of the Board of Trade. The same applies to the Germanischer Lloyd, Lloyd's Register, and the Bureau Veritas.

§ 59.

Launch Boilers.

1) **I. Classification.** Steam launches are carried not only by all war-ships but also by many of the larger passenger steamers. The boilers must of course be as powerful as possible for their weight and too much water is inadmissible. The tendency is therefore to adopt a very high pressure and to keep down the thickness of cylindrical shell-plating by using small diameters. Launch boilers are either Tubular or Tubulous. Classification.

2) **II. Tubular launch boilers** exactly resemble those for ships, but Scotch boilers are seldom met with because of their great height. Most launches have Tubular boilers.

a) Navy-type boilers, or
b) Locomotive boilers.

3) a. **The navy-type boilers** are in general the same as that shewn in Figs. 1 to 3, Pl. 37. For the higher pressures they are built of steel and often have a comparatively lofty dome in order to produce drier steam. Two eyes are usually riveted on each end for convenience of slinging when the boiler is taken out of the boat for overhauling at sea. The boiler is fastened by four clamps passing over bolts and secured by forelocks, so that it is very simply shipped and unshipped. It can be cleaned by taking off the top of the dome and removing the hand-hole lids and cleaning plugs in the end plates. Navy-type boilers.

4) b. **The locomotive boilers** do not generally differ much from those represented in Pl. 28, they are only of smaller dimensions and have therefore lighter plating and stays: Figs. 4 to 6, Pl. 37 shew a very useful form of locomotive boiler for launches. The fire-box takes the shape of a vertical cylinder to which the barrel is united. This design requires no staying except for the fire-box crown, rendering the boiler comparatively light. The bridge is supported by a cross-tube passing through the furnace and accessible from the outside through two mud-holes. The boiler can be cleaned by means of a man-hole in the crown of the fire-box shell. To save space in the boat, the fire door is at the side. Locomotive Boilers.

5) **III. Water-tube boilers for launches** are almost as various in design as those for ships and many of the types described in § 57, for instance BELLEVILLE'S and THORNYCROFT'S, are used in the same manner both for ships and launches. Only such boilers are therefore referred to here as are intended for launches exclusively. They are for the most part straight-tubed water-chamber boilers, to Classification

which class the first three in the following list belong, the other two are curved-tubed boilers.

a) HOLTZ'S boiler,
b) PENELLE'S boiler,
c) TOWNE'S boiler.
d) HERRESHOFF'S boiler,
e) WARD'S boiler.

Holtz's.

6) a. **Holtz's boiler,** Figs. 7 to 9, Pl. 37 consists of several rows of tubes fitted zigzag into a water-chamber at each of their ends. The water-chambers communicate through oblong openings at the top with a top drum of elliptical section which carries a dome. The fire-door is at the side, the gases travel over the tubes and top drum to the funnel at the back. The tubes can be swept by means of doors in the sides of the casing. Boilers of similar design are very much used, among others by DES VIGNES & Co. of Teddington for their small fast launches.

Penelle's.

7) b. **Penelle's boiler,** *) Figs. 3 and 4, Pl. 38, adopted in the French Navy, contains a nest of tubes and a cylindrical top drum bounded at each end by a water-chamber of convex form which can be removed for getting at the inside of the tubes. The connecting bolts are so numerous and large that the stress per sq. mm of section is only 2.7 kg. At each side of the top drum there is a cylindrical steam-collector attached by necks to the drum and the steam-pipe is branched to both collectors, between which the funnel is placed. These boilers are reported to have done well.

Towne's.

8) c. **Towne's boiler,** Figs. 8 to 10, Pl. 38 also belongs to the water-tube boilers with water-chambers and top-drum. In this case the water-chambers are united into a hollow rectangular double casing surrounding the whole boiler. A series of tubes connect the top drum and the water-chamber, expanded at their ends as usual, the plating of the top drum being doubled where it takes them. The water-tubes are arranged in parallel rows crossing each other, the internal plates of the lateral portions of the water-chamber being bent so as to take the tube ends square and opposite each end of each tube a hole is provided, closed with a conical plug (Fig. 10) and copper ring about 1.5 mm thick. The casing on the top consists of two thicknesses of sheet steel with slag-wool or magnesia brick between them. Outside the casing at the back there is a down-comer of large diameter connecting the top drum by a branch with the bottom of each of the water-chambers. To increase the circulation still further the feed-pipe is taken into the top drum and terminates at the

*) A. F. N. Bienaymé. Les machines marines. Paris 1887, Pl. 152.

orifice of the down-comer in a nozzle of 2.5 mm bore Above the water-tubes there is a feed-heater on each side of the boiler consisting of several tubes among which the hot gases pass. The TOWNE boiler was adopted for launches in the U. S. navy, but being only moderately successful, was displaced by the WARD boiler.

9) d. **Herreshoff's boiler,** Figs. 1 and 2, Pl. 38 was accepted in the British and U. S. navies because of its extraordinary rapidity of getting up steam and is distinguished for its light weight and capacity to bear heavy pressures. It consists of two long tubes coiled upon each other. They are formed by welding a number of tubes together at the ends. The outside coil is cylindrical and the tube of constant diameter, the inner one bell-shaped and the tube of smaller section at the crown of the bell than at the sides. The top ends of the inner and outer tubes are united, the bottom end of the outer one connects to the delivery pipe of a special feed-pump, and the free end of the inside tube leads to a steam-trap whence the steam is taken to the engine. The interior of the bell forms the furnace which is unusually roomy and favourable to good combustion. The lower part round the grate is lined with fire-brick. The cylindrical portion of the boiler is lagged with two or three thicknesses of sheet with air-spaces between them. The crown is covered with slag-wool. The turns in the sides of the outer coil and the crown of the inner one are kept close together but they have small spaces between them in the sides of the inner and the crown of the outer one, so that the gases can get at the outer coil and from the space between the two coils up into the funnel. Herreshoff's.

10) The water, forced by the special pump into the outer coil which acts as a feed-heater, passes gradually upwards to the boiler crown and into the inner coil. Thence it proceeds, almost in a state of steam, into the wide lower portion of the inner coil where the evaporation is accelerated. If so little feed-water enters the boiler that it is already evaporated in the upper part of the inner coil, the steam in the lower part of this must be greatly superheated and injure the cylinders &c. of the engine. HERRESHOFF prevents this superheating by pumping more water through the boiler than can evaporate in it. The steam trap therefore always contains a mixture of steam and water of from 15 to 20 % moisture. This trap which serves the two-fold purpose of arresting the suspended water and steadying the steam pressure as a dome, is a specially good point of the HERRESHOFF boiler. The mixture entering the trap from the Circulation.

pipe *A* is separated into steam and water, the former passes on to the engine by the wrought iron pipe *B* in which the stop-valve is fitted, the latter falls to the bottom. The pipe *C*, also of wrought iron, leads to the safety-valves.

Special feed-pump.

11) The gauge-glass shews the height of the water which is in excess of the quantities evaporated and forced through the boiler. The speed is so high that all deposits of grease and other impurities contained in the feed are washed out into the trap whence they are afterwards expelled by blow-off cocks at the bottom (*D*) and surface (*E*). The water remaining in the trap after blowing off passes through the pipe F into the suction-box of the special feed-pump. The pump is in connection, through a pipe which can be shut off, with the condenser, so as to afford a ready means of disposing of large quantities of water suddenly forced through the boiler. The gauge-glass on the trap indicates a flooding of this sort as well as a deficiency of the superabundant water which ought to flow through the boiler. When such a deficiency occurs, the supplementary feed must be increased so that the main-engine feed-pump can supply more water from the hot-well to the suction tank of the special boiler-pump. As the pipe *F* from the steam-trap also leads into this suction tank which is closed and stands under the same pressure as the steam-trap, it follows that the bucket of the boiler-pump is nearly in equilibrio, having little more to overcome than the frictional resistance of the water in the tubes composing the boiler. The air-pump takes the necessary supplementary feed from a fresh-water tank and conveys it to the hotwell the overflow pipe of which leads to the suction tank of the boiler pump.

Trial-trip Results.

12) The first contractor's trial in the Thames of the HERRESHOFF boat in December 1878 was satisfactory inasmuch as the required speed of 16 knots was attained. But the trials, held in the Solent in 1879*) to demonstrate the serviceable qualities of the boiler, were anything but favourable. It is true the badly-designed condenser and not the boiler was in fault. In 1881**) the British Admiralty made some comparative trials of steam launches of various sizes by WHITE and HERRESHOFF in which the HERRESHOFF boats with the boiler just described always shewed themselves the faster. The principal reason of this was the small weight of their boilers which when charged were only half the weight of WHITE'S Navy-type boilers. The small weight of water allowed of such rapid heating up that steam could be raised to 4 atmos. in 5 min. 50 sec.

*) Engineering. 1879. I. p. 305.

**) Transactions of the Institution of Naval Architects. London 1887. P. 328.

13) But on the other hand the small quantity of water necessitates extremely careful firing if the steam is to be kept steady. A further objection is that when the engines are stopped the whole contents of the boiler are quickly evaporated and the overheating of the inner coil can only be prevented with considerable difficulty. That HERRESHOFF boilers are more rapidly worn out than cylindrical ones Engineer LEONARD, U. S. N. has demonstrated*) from his experience with three of these boilers, only one of which remained fit for work two years. They may therefore be now considered obsolete, — at any rate they are no longer fitted to war-ships. Drawbacks of the Herreshoff boiler.

14) e. **The Ward Boiler,** Figs. 5 to 7, Pl. 38, was subjected to a thorough trial in 1884 by Chief Engineers ISHERWOOD, ZELLER, and HUNT, U. S. N.**) and afterwards introduced into the service for launches with hitherto completely satisfactory results. It is intended for a high working pressure and is tested to 35 atmos. with water. During the water-test and 13 steam trials above referred to of about 24 hours each, the working pressure being as high as 18.3 atmos., neither leaks nor any sign of deformation nor yielding of fastenings appeared. The following are the essential points of the design. Above the ash-pit there is a horizontal cast steel ring of hollow circular section provided on its inner periphery with radial lugs carrying a wrought iron ring which receives the cast iron fire-bars. On the outside of the annular tube there is a rib of horizontal T section to which the casing of the boiler and ash-pit are attached by screws. The annular tube is filled with water so that one fourth of its surface may be regarded as heating surface while the remainder is exposed to the cooling current of air entering the ash-pit. The annular tube is interrupted at the fire-door and diverted upwards in the form of a rectangle so as to surround the door. Entirely round the upper part of the annular tube there are fitted two rows of sockets placed zigzag to which the upright tubes surrounding the fire are connected by unions. The upper ends of the tubes are bent in such a manner as to bring them into four horizontal rows in zigzag order and similarly connected by unions to a vertical cast steel cylinder. The bottom of this cylinder is slightly dished and carries three circular rows of suspended tubes jointed to it with unions and closed with caps at their lower ends which project into the fire. The mouth of each suspended tube is closed Ward Boilers.

*) Journal of the American Society of Naval Engineers. Washington. 1890. Pl. 168.

**) Report made to the bureau of steam engineering Sept. 10, 1884 by a board of United States Naval Engineers, on the steam boiler invented by Ch. Ward. Printed by the authority of U. S. Navy Department.

Particulars of 40 various Marine

№	Ship's Name	Plate referred to	Year constructed	Constructor	Working Pressure in atmos.	No. of Boilers	IHP of Engines: Designed	IHP of Engines: Natural Draught	IHP of Engines: Artificial Draught
1	2	3	4	5	6	7	8	9	10

I. Box

№	Ship's Name	Plate referred to	Year constructed	Constructor	Working Pressure in atmos.	No. of Boilers	Designed	Natural Draught	Artificial Draught
1	Despatch Boat "Kaiseradler"	Pl. 19 Figs. 1 & 2	1876	Germania	2.0	2 4	3000	2170	205[illegible]
2	Cruiser "Blücher"	Pl. 19 Figs. 3 & 4	1877	Germania	2.0	4	2500	2055	252[illegible]
3	„ "Stein"	„	1880	Vulcan	2.5	4	2500	2151	270[illegible]
4	Armour-clad "Preussen"	„	1875	Vulcan	2.0	6	5400	3861	43[illegible]
5	„ "Friedrich d. Grosse" . . .	„	1876	Germania	2.0	6	5400	4446	49[illegible]
6	„ "Kaiser"	„	1873	Penn	2.0	8	8000	5200	78[illegible]

II. Navy

№	Ship's Name	Plate referred to	Year constructed	Constructor	Working Pressure in atmos.	No. of Boilers	Designed	Natural Draught	Artificial Draught
7	Despatch Boat "Zieten"	Pl. 20 Figs. 3 & 4	1875	Penn	5.5	6	2350	1951	235[illegible]
8	Cruiser "Habicht"	„	1879	Schichau	6.0	2	600	698	875
9	„ "Sperber"	„	1888	I. D. Wilhelmshav.	7.0	4	1500	1509	171[illegible]
10	* „ "Falke"	Pl. 21 Figs. 3 & 4	1891	I. D. Kiel	12.0	4	2800	2205	291
11	First Class Launch	Pl. 37 Figs. 1 to 3	1894	I. D. Kiel	9.0	1	40	—	44
12	Third „ „	„	1892	I. D. Kiel	9.0	1	20	—	21

IV. Single-ended

№	Ship's Name	Plate referred to	Year constructed	Constructor	Working Pressure in atmos.	No. of Boilers	Designed	Natural Draught	Artificial Draught
13	Armour-clad "Wespe"	Pl. 22 Figs. 1 & 2	1878	I. D. Kiel	4.0	4	700	660	8[illegible]
14	Despatch Boat "Blitz"	„	1882	I. D. Wilhelmshav.	5.0	8	2700	2273	28[illegible]
15	Cruiser "Olga"	„	1880	Vulcan	5.0	8	2100	2121	220[illegible]
16	„ "Sophie"	„	1881	Germania	5.0	6	2100	1930	225[illegible]
17	„ "Alexandrine"	„	1886	I. D. Kiel	5.0	8	2400	1974	255[illegible]
18	Armour-clad "Oldenburg"	Pl. 24 Figs. 3 & 4	1886	Vulcan	5.0	8	3900	3236	4[illegible]4
19	* „ "Brandenburg"	„	1891	Vulcan	12.0	12	9000	8105	9997
20	* „ "Wörth"	„	1892	Germania	12.0	12	9000	8199	10303
1	2	3	4	5	6	7	8	9	10

*) Boilers marked thus are of steel, the rest of iron.

Boilers in the German Navy.

Principal Dimensions			Plating					Staying		Furnaces			Grate				Heating Surface in each Boiler		
Lenght	Breadth or Diameter	Height	Shell	Bottom	Furnaces	Tube-plates	Uptake	Diar. of the Longitudinal stays	Diar. of the C. chr. stays	No. in each boiler	for Box and Locomotive Boilers Height / Diameter of circular Furnaces	for Box and Locomotive Boilers Breadth	Length per Furnace	Breadth in each Furnace	Total surface in each Boiler	Clear surface in each Boiler	Tubes	Other Parts	Total
m	m	m	mm	mm	mm	mm	mm	mm	mm		m	m	m	m	sqm	sqm	sqm	sqm	sqm
11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
Boilers.																			
.819	6.199	3.160	10.0	13	11	16	13	36	32	5	1.020	1.046	1.674	1.046	8.753	2.000	180.80	50.47	231.38
„	3.779	„	„	„	„	„	„	„	„	3	„	„	„	1.044	5.253	1.200	108.48	30.33	138.81
.812	6.199	3.160	10.0	13	11	16	13	36	32	5	1.150	1.046	1.674	1.046	8.755	2.233	183.12	47.43	230.55
.820	6.407	3.148	10.0	13	11	16	13	37	33	5	1 150	1.026	1.674	1.046	8.756	2.233	181.60	58.12	239.70
.060	5.492	3 766	10.0	13	11	16	13	39	33	5	1.098	0.940	2.295	0.922	10.58	3.174	234.14	52.40	287.33
.080	5.492	3.770	10.0	13	13	16	13	45	32	5	1.120	0.900	2.336	0.90	10.511	2.622	215.81	59.28	275.09
.067	5.20	3.680	10.0	13	10	16	13	39	32	5	1.143	0.860	2.438	0.824	10.230	2.771	221.71	57.17	275.24
Boilers.																			
.283	2.261	14.0	14	10	18	6	50 38	32	2		0.927		1.980	0.927	3.670	1.160	104.698	14.60	119.30
480	2.400	13.0	16	15	21	6	33	32	2		0.970		2.520	0.970	4.888	1.530	135.68	18.12	153.80
.700	2.380	14.5	18	12	19	4	54	32	2		0.960		1.700	0.960	3.250	0.810	100.50	15.50	116.00
020	2.840	22.0	23	15	22	5	59	35	3		0.970		1.880	0.930	5.250	2.410	156.50	1350	170.00
450	1.110	10.0	12	10	12	2	36 22	26	1		0.695		0.660	0.860	0.570	0.280	15.16	16.40	16.80
170	0.856	8.0	11	9	11	1,5	30 & 20	20	1		0.510		0.510	0.570	0.322	0.160	6.17	0.95	7.12
cotch Boilers.																			
350	2.816	18 & 16.0	18 & 16	11	18	5	48	25	2		0.800		1.660	0.800	2.650	0.645	61.85	11.80	73.65
900	2.960	16.0	16	11	18	5	45	32	2		0.940 0.840			0	3.450	0.789	84.59	13.91	98.50
600	3.150	18.0	15	12	18	8	52	32	2		0.920			0	3.400	1.015	80.00	18.00	98.00
700	3.300	16.0	16	12	16	8	40	32	2		1.030			.3	3.800	1.040	90.40	15.60	106.00
600	3.200	18.0	15 & 12	10	18	6	52	32	2		0.920			0	3.680	0.650	81.35	17.42	98.80
000	3.840	20.0	18	11	18	6	55 43	32	3		0.960			0	5.760	0.780	135.50	26.50	162.00
020	3.960	28.5	22	14	22	4,5	58 51	38	3		0.950			0	5.830	1.032	162.70	28.20	190.90
900	3.960	28.5	22	14,5	22	5	57 46	36	3		0.950			0	5.850	2.260	166.50	29.50	196.00
11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30

№	Ship's Name	Plate referred to	Tubes: External Diameter mm	Tubes: Thickness mm	Tubes: Length between Tube plates m	Tubes: Number	Funnel: Height above Bars m	Funnel: Sectional Area, Greatest sqm	Funnel: Sectional Area, Smallest sqm	Section for Draught in each Boiler: Ash-pit sqm	Section for Draught in each Boiler: Over Bridge sqm	Section for Draught in each Boiler: Through Tubes sqm	Section for Draught in each Boiler: Uptake sqm
			31	32	33	34	35	36	37	38	39	40	41
	I. Boi												
1	Despatch Boat "Kaiseradler"	Pl. 19 Figs. 1 & 2	63.5	2.5	1.700	530	14.64	24.10		1.743	1.435	1.420	[illegible]
			"	"	"	318	"	"		1.045	0.861	0.852	[illegible]
2	Cruiser "Blücher"	Pl. 19 Figs. 3 & 4	63.5	2.5	1.732	530	17.45	4.773	4.503	2.370	1.930	1.424	[illegible]
3	" "Stein"	"	76.0	3.0	1.770	430	12.69	4.948	4.752	2.262	1.845	1.654	[illegible]
4	Armour-clad "Preussen"	"	63.5	2.5	2.014	572	18.25	7.20		2.155	29.00	1.550	[illegible]
5	" "Friedrich d. Grosse"	"	63.5	2.5	1.900	580	20.59	7.916		1.580	28.47	1.532	[illegible]
6	" "Kaiser"	"	76.0	3.0	1.950	475	20.04	2 × 4.091		1.956	19.52	1.828	[illegible]
	II. Nav												
7	Despatch Boat "Zieten"	Pl. 20 Figs. 3 & 4	70.0	3.0	2.110	226	12.50	24.81		0.410	0.610	0.720	[illegible]
8	Cruiser "Habicht"	"	76.0	3.0	2.120	268	13.00	2.000		0.560	0.480	1.030	[illegible]
9	" "Sperber"	"	63.5	2.75	1.800	246	13.00	1.540		0.470	0.630	0.720	[illegible]
10 *	" "Falke"	Pl. 21 Figs. 3 & 4	63.5	4.0	1.990	324	13.40	2.540		0.850	1.124	0.841	[illegible]
11	First Class Launch	Pl. 37 Figs 1 to 3	35.0	2.0	0.625	234	3.00	0.071		0.110	0.280	0.176	[illegible]
12	Third " "	"	30.0	2.0	0.504	130	2.62	0.041		0.040	0.130	0.070	[illegible]
	III. Single-en												
13	Armour-clad "Wespe"	Pl. 22 Figs. 1 & 2	63.5	2.5	1.700	181	12.20	1.605		0.477	0.430	0.478	[illegible]
14	Despatch Boat "Blitz"	"	76.0	3.0	2.160	164	12.00	1.728		0.546	0.500	0.631	[illegible]
15	Cruiser "Olga"	"	76.0	3.0	1.870	180	14.70	2.296	2.138	0.480	0.664	0.695	[illegible]
16	" "Sophie"	"	76.0	3.0	1.930	198	13.16	3.801	2.950	0.824	0.800	0.717	[illegible]
17	" "Alexandrine"	"	76.0	3.0	1.920	182	14.50	1.910	1.770	0.540	0.610	0.700	[illegible]
18	Armour-clad "Oldenburg"	Pl. 24 Figs. 3 & 4	76.0	3.0	2.200	258	19.00	2.830		0.78	0.800	0.960	[illegible]
19 *	" "Brandenburg"	"	63.5	3.0	2.075	306	20.10	6.160	4.150	0.830	1.060	0.960	[illegible]
20 *	" "Wörth"	"	63.5	3.0	2.135	319	18.80	7.070	4.150	1.020	0.940	0.900	[illegible]
			31	32	33	34	35	36	37	38	39	40	

Contents of each Boiler: Steam-space (cbm)	Contents of each Boiler: Water-space (cbm)	Proportions: Steam-space to Water-space	Proportions: Heating Surface per IHP	Proportions: Grate Surface per IHP	To Grate-area: Heating Surface	To Grate-area: Area through Ash-pit	To Grate-area: Area over Bridge	To Grate-area: Area through Tubes	To Grate-area: Sectional area of Funnel	To Grate-area: Clear grate area	Weight: per Boiler (kg)	Weight: per Boiler with water (kg)	Weight: of all Boilers per IHP (kg)	Weight: of total water per IHP (kg)	Weight: of all Boilers with water per IHP (kg)	Performance: IHP per sqm of Grate	Performance: IHP per ton total Weight of Boilers with water	Remarks
42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60
Boilers.																		
11.934	19.033	0.56	0.34	0.013	26.5	0.20	0.16	0.16	0.12	0.23	25000	44033	38.2	28.3	66.5	77.9	15.1	Large } Boiler.
7.430	11.690	"	"	"	"	"	"	"	"	"	16200	27890						Small
11.370	17.690	0.64	0.37	0.014	26.3	0.27	0.22	0.16	0.13	0.25	24480	42170	39.1	28.3	67.4	71.4	14.8	
11.600	20.300	0.57	0.38	0.014	27.4	0.25	0.21	0.18	0.14	0.25	26416	46716	42.2	32.5	74.7	71.4	14.0	
13.810	22.70	0.61	0.32	0.012	27.1	0.20	0.27	0.15	0.11	0.30	28570	51270	31.8	25.2	57.0	85.0	17.5	
14.790	22.811	0.65	0.30	0.012	26.5	0.15	0.27	0.15	0.12	0.25	27916	50727	310	25.3	56.3	85.6	17.3	
11.951	20.104	0.59	0.27	0.010	26.8	0.19	0.15	0.17	0.10	0.27	26889	46988	26.9	20.1	47.0	07.8	21.3	
Boilers.																		
4.130	9.565	0.30	0.30	0.009	32.5	0.11	0.16	0.19	0.11	0.31	12939	22504	33.0	24.3	27.3	106.7	17.4	
4.640	10.591	0.44	0.37	0.011	31.5	0.12	0.12	0.15	0.20	0.31	16492	27083	54.9	35.3	90.2	61.4	11.1	
4.000	8.140	0.48	0.31	0.010	35.6	0.14	0.19	0.22	0.12	0.25	14877	23017	39.7	21.7	61.4	115.4	15.5	
4.850	12.950	0.37	0.24	0.007	32.4	0.162	0.21	0.16	0.07	0.46	25469	38419	36.5	18.5	54.9	133.3	17.9	
0.330	0.570	0.58	0.42	0.014	29.5	0.19	0.51	0.31	0.12	0.50	1528	2098	38.2	14.5	52.6	70.2	19.1	
0.150	0.280	0.54	0.36	0.020	22.1	0.11	0.40	0.21	0.13	0.49	726	1266	36.2	27.0	63.2	62.5	15.8	
cotch Boilers.																		
3.420	6.150	0.40	0.46	0.015	27.8	0.18	0.16	0.18	0.15	0.24	8679	14829	49.6	35.1	84.7	66.0	11.8	
3.640	8.962	0.40	0.36	0.012	28.5	0.16	0.14	0.18	0.20	0.25	11550	20512	42.0	32.6	74.6	98.0	16.5	
4.250	8.087	0.57	0.37	0.013	27.0	0.14	0.19	0.20	0.17	0.26	12787	20874	48.7	30.8	79,5	77.2	12.6	
4,350	10.710	0.40	0.30	0.011	28.0	0.22	0.21	0.18	0.17	0.27	12922	23632	36.9	30.6	67.5	92.1	14.8	
4.880	8.350	0.58	0.33	0.012	26.8	0.147	0.16	0.19	0.15	0.18	13071	21421	43.6	27.8	71.4	81.5	14.0	
7.260	14.050	0.52	0.20	0.170	25.0	0.127	0.14	0.17	0.19	0.14	19996	34046	41.0	28.7	69.7	84.6	14.3	
5.110	15.680	0.32	0.25	0.008	32.8	0.143	0.18	0.16	0.91	0.18	31700	47380	42.3	20.9	63.2	128.6	15.8	
6.370	14.830	0.43	0.26	0.008	33.5	0.174	0.16	0.17	0.93	0.38	31270	56100	41.7	19.7	74.8	128.2	13.2	
42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60

№	Ship's Name	Plate referred to	Year constructed	Constructor	Working Pressure in atmos.	No. of Boilers	IHP of Engines: Designed	IHP of Engines: Natural Draught	IHP of Engines: Artificial Draught
1	2	3	4	5	6	7	8	9	10
	IV. Double-ended								
21	Despatch Boat "Greif"	Pl. 26 Figs. 3 & 4	1886	I. D. Kiel	7.0	6	5400	4108	[illegible]
22	Cruiser "Irene"	"	1888	Vulcan	7.0	4	8000	6649	[illegible]
23	" "Prinzess Wilhelm"	Pl. 27 Figs. 3 & 4	1889	Germania	7.0	4	8000	7919	[illegible]
24	*Transport "Pelikan"	Pl. 26 Figs. 3 & 4	1891	I. D. Wilhelmshav.	12.0	2	3000	1973	[illegible]
25	*H. M. Yacht "Hohenzollern"	Pl. 27 Figs. 3 & 4	1892	Vulcan	12.0	4 4	9000	6571	[illegible]
26	*Cruiser "Kaiserin Augusta"	Pl. 26 Figs. 3 & 4	1892	Germania	12.0	8	12000	12506	[illegible]
27	* " "Gefion"	"	1893	Schichau	12.0	4 2	9000	6538	[illegible]
	V. Locomotive								
28	Amour-clad "Brummer"	Pl. 28 Figs. 1 & 2	1883	Weser	7.0	2	1500	1181	[illegible]
29	Despatch Boat "Jagd"	"	1888	Weser	10.0	4	4000	2580	[illegible]
30	*Amour-clad "Siegfried"	"	1890	Germania	12.0	4	4800	3629	[illegible]
31	* " "Hagen"	Pl. 29 Figs. 5 & 6	1894	I. D. Kiel	12.0	4	4500	3322	[illegible]
32	*Despatch Boat "Comet"	Pl. 28 Figs. 1 & 2	1892	Vulcan	12.0	4	4500	—	[illegible]
33	*Launch Class A	Pl. 29 Figs. 5 & 6	1894	Germania	12.0	1	150	--	[illegible]
34	* " " B	"	1894	Germania	12.0	1	120	—	[illegible]
	VI. Water-								
35	Dock-yard Pinnace (captured French) . . .	Pl. 30 Figs. 1 & 2	1875	I. D. Kiel	6.0	1	20	—	—
36	Tender "Ulan" (old boiler)	"	1876	Löwe, Berlin	8.0	4	800	275	—
37	*Tug "Föhn"	Pl. 31 Figs. 1 & 2	1890	I. D. Kiel	10.0	1	150	—	234
38	*Transport "Rhein"	Pl. 30 Figs. 13 & 14	1894	Dürr, Ratingen	14.0	1	340	290	[illegible]
	VII. Auxiliary								
39	Auxiliary boilers of large cruisers	Pl 35. Figs. 9 & 10	1877	I. D. Kiel	2.0	1	—	—	—
40	" " " small "	"	1877	I. D. Kiel	2.0	1	—	—	—
1	2	3	4	5	6	7	8	9	10

Principal Dimensions: Length	Principal Dimensions: Breadth or Diameter	Principal Dimensions: Height	Plating: Shell	Plating: Bottom	Plating: Furnaces	Plating: Tube-plates	Plating: Uptake	Saying: Diar. of the Longitudinal stays	Saying: Diar. of the C. chr. stays	Furnaces: No. in each Boiler	Furnaces: Height — for Box and Locomotive Boilers Diameter of circular Furnaces	Furnaces: Breadth	Grate: Length per Furnace	Grate: Breadth in each Furnace	Grate: Total surface in each Boiler	Grate: Clear surface in each Boiler	Heating Surface in each Boiler: Tubes	Heating Surface in each Boiler: Total	Heating Surface in each Boiler: Other Parts
m	m	m	mm	mm	mm	mm	mm	mm	mm		mm	mm	m	m	sqm	sqm	sqm	sqm	sqm
11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
cotch Boilers.																			
500	3.200		18	18	11	19 18	8	54	32	4	0.920		2.000	0.920	7.36	2.150	177.64	29.86	207.50
000	4.510		25	25	13	22 20	8	55	32	6	1.250		2.000	1.250	15.00	4.080	367.45	68.55	436.00
.000	4.510		25	18		12,5	5	50	32	8	1.050		1.905	0.980	16.00	4.450	399.30	75.70	475.00
.900	4.000		30	15	15	25 22	4	59	35	6	1.000		2.000	1.000	12.00	5.740	320.4	69.60	390.00
.150 .270	4.200		30	22 ob. 18 u.	13,5	22 v. 20 h.	5	58	42	8 4	1.000		2.000	1.000	16.00 18.00	2.260	384.50 192.20	62.00 31.00	446.50 223.20
.900	4.000		30	22	15	22	5 4	58 46	32	6	1.000		2.000	1.000	12.00	4.000	350.00	61.50	411.50
.00	3.850 3.450		28 25	25 25	16 15	22 22	5 4	60 45	30 25	6 4	1.150 1.000		2.100	1.150 0.950	12.00 9.20	3.490 2.660	325.00 257.30	54.00 39.50	379.00 296.80
oilers.																			
478 382	2.304 3.522	2.217	19	19	15	25 22	5	49 32	26	2	1.500	0.993	1.900	0.993	3.78	0.980	240.00	20.67	260.67
660 340	2.304 2.314	2.422 —	14 18	22 19	16	25	5	49	32	2	1.690	0.993	2.000	0.950	3.80	2.300	237.85	17.25	255.10
600 948	3.030 2.350	1.896 —	16 18	22.5 20.5	16	25 20	5 4	57	32	2	1.880	1.350	2.000	1.350	5.40	1.430	229.70	22.60	252.30
110 780	3.300 2.740	1.767 —	12 18	20	17	22 20	5 4	60 48	35 28	2	1.270	2,00	1.990	1.440	5.72	2.270	221.30	25.00	246.30
930 310	3.000 2.500	1.717 —	22 18	18.5	17	22 20	3	49 46	35	2	1.700	1.290	2.100	1.290	4.28	2.10	193.60	19.00	212.60
280 820	1.200 1.118	0.788 —	9	11	7	12	2	32	32 22	1	0.650	1.10	0.930	1.070	1.00	1.370	22.80	5.05	27.85
10 760	1.200 1.118	0.788 —	9	11	7	12	2	32	22	1	0.650	1.050	0.780	1.070	0.85	1.160	20.71	3.29	24.00
ıbe Boilers.																			
000	0.660	1.400	—	—	—	—	—	—	—	1	—	—	0.775	0.521	0.403	0.137	2.70	0.40	3.10
670	2.050	2.700	—	—	—	—	—	—	—	2	—	—	2.130	0.705	3.000	0.725	78.12	9.02	87.14
180	1.830	2.650	8	13	—	15	2.5	—	24	2	—	—	1.700	1460	2.500	1.000	87.50	1.20	88.70
500	2.750	4.160	22	18	—	18	2.0	50	38	3	—	—	1.805	0630	3.560	1.580	88.50	9.13	97.63
oilers.																			
—	1.080	2.690	10	11	11	16	10	—	—	1	—	—	0.900		0.636	0.156	11.07	5.33	16.40
—	0.820	2.310	8	10	10	12	8	—	—	1	—	—	0.650		0.232	0.066	4.41	2.65	7.06
11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30

No	Ship's Name	Plate referred to	Tubes: External Diameter mm	Tubes: Thickness of Plating mm	Tubes: Length between Tube-plates m	Tubes: Number	Funnel: Height above Bars m	Funnel: Sectional Area, Greatest sqm	Funnel: Sectional Area, Smallest sqm	Section for Draught in each Boiler: Ash-pit sqm	Over Bridge sqm	Through Tubes sqm	Uptake sqm
			31	32	33	34	35	36	37	38	39	40	41
	IV. Double-end…												
21	Despatch Boat "Greif"	Pl. 26 Figs. 3 & 4	76.0	3.0	2.000	308	14.60	1.830		1.090	1.040	1.050	0.920
22	Cruiser "Irene"	"	76.0	3.0	2.250	684	16.20	5.090	3.140	3.003	2.972	2.477	[illegible]
23	" "Prinzess Wilhelm"	Pl. 27 Figs. 3 & 4	76.0	3.0	2.200	760	16.88	5,090	3.140	2.880	2.622	2.746	[illegible]
24	*Transport "Pelikan"	Pl. 26 Figs. 3 & 4	76.0	3.0	2.150	480	16.85	2.830		2.080	1.030	1.120	[illegible]
25	*H. M. Yacht "Hohenzollern"	Pl. 27 Figs. 3 & 4	76.0	3.5	2.250	580 290	20.67	6.470	4.050	1.260	1.420	1.300	[illegible]
26	*Cruiser "Augusta"	Pl. 26 Figs. 3 & 4	63.5	3.0	2.912	636	16.60	8.000	3.100	2.340	1.950	2.040	[illegible]
27	* " "Gefion"	"	64.0	3.0	2.320	552 448	18.80	5.720 4.710	2.540 2.140	2.300 2.000	1.630 1.500	3.580 2.930	[illegible]
	V. Locomoti…												
28	Armour-clad "Brummer"	Pl. 28 Figs. 1 & 2	52.0	2.5	3.500	420	10.38	1,327		0.690	2.340	0.730	[illegible]
29	Despatch Boat "Jagd"	"	52.0	3.0	3.500	416	12.00	2.83	1.33	0.850	1.190	0.670	[illegible]
30	*Amour-clad "Siegfried"	"	52.0	3.0	3.080	464	18.85	3-300		0.980	1.450	0.660	[illegible]
31	* " "Hagen"	Pl. 29 Figs. 5 & 6	55.0	2.5	2.900	486	18.80	3.560		0.780	1.960	0.770	[illegible]
32	*Despatch Boat "Comet"	Pl. 28 Figs. 1 & 2	50.0	2.5	2.400	514	13.06	1.330		0.850	1.200	0.817	[illegible]
33	*Launch Class A	Pl. 29 Figs. 5 & 6	32.0	1.7	0.850	268	3.90	0.148		0.370	0.440	0.171	[illegible]
34	* " " B	"	32.0	1.7	0.780	265	3.75	0.125		0.330	0.440	0.171	[illegible]
	VI. Water…												
35	Dock-yard Pinnace (captured French)	Pl. 30 Figs. 1 & 2	79.0	5.0	0.693	36	3.45	0.039	—	0.089	—	—	[illegible]
36	Tender "Ulan" (old boilers)	"	100.0	6.0	2.310	126	13.20	0.785	—	0.700	—	—	1.400
37	*Tug "Föhn"	Pl. 31 Figs. 1 & 2	65.0	2.7	1,800	260	6.30	0.380	—	1.163	—	—	0.380
38	*Transport "Rhein"	Pl. 30 Figs. 13 & 14	115.0	4.5	2.150	118	9.20	0.710	—	0.774	—	—	0.640
	VII. Auxiliar…												
39	Auxiliary boilers of large cruisers	Pl. 30 Figs. 9 & 10	37.0	2.5	0.930	124	—	0.0615		0,180	—	0 490	[illegible]
40	" " " small "	"	37.0	2.5	0.790	54	—	0.0433		0.991	—	0.234	[illegible]
			31	32	33	34	35	36	37	38	39	40	41

Contents of each Boiler		Proportions										Weight					Performance		Remarks
Steam-space	Water-space	Steam-space to Water-space	Heating surface per IHP	Grate surface per IHP	Heating surface	Area through Ash-pit	Area over Bridge	Area through Tubes	Sectional area of Funnel	Clear grate area	per Boiler	per Boiler with water	of all Boilers per IHP	of total water per IHP	of all Boilers with water per IHP	IHP per sqm of Grate	IHP per ton total Weight of Boilers with water	
cbm	cbm				To Grate-area						kg	kg	kg	kg	kg			
42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60
cotch Boilers.																		
9.730	17.370	0.56	0.23	0.008	28.2	0.148	0.14	0.14	0.12	0.29	23954	41324	2.66	19.3	45.9	122.3	21.8	
9.75	35.000	0.56	0.22	0.008	29.0	.0200	0.20	0.17	0.10	0.27	57000	92000	28.5	21.2	46.0	133.3	21.7	
7.76	36.750	0.48	0.23	0.008	29.7	0.180	0.16	0.17	0.07	0.28	57873	94623	28.9	18.0	47.3	125.0	21.3	
3.340	29.730	0.45	0.26	0.008	32.5	0.173	0.17	0.18	0.23	0.48	55793	85523	38.2	19.8	57.0	125.0	17.5	
4.250 7.250	31.950 16.850	0.44	0.29	0.010	27.9	0.160	0.18	0.16	0.34	0.14	58539 33041	90489 49891	37.4	21.7	62.4	151.1	20.5	Double-ended Boilers. Single-ended Boilers.
3.800	31.000	0.44	0.27	0.008	34.3	0.195	0.16	0.17	0.41	0.33	62113	93113	41.4	20.7	62.1	125.0	16.1	
1.800 9.600	30.200 23.500	0.40	0.23	0.007	32.0	0.291 0.217	0.16	0.31	0.17	0.29	47512 38193	77712 63693	29.6	18.7	48.2	136.3	20.7	Large } Boiler. Small }
Boilers.																		
5.500	12.910	0.43	0.17	0,005	70.0	0.180	0.62	0.19	0.17	0,26	21600	31600	28.8	17.2	42.1	198.4	23.7	
2.900	4.800	0.60	0.25	0.002	67.1	0.220	0.31	0.18	0.10	0.60	21641	26441	21.6	4.8	26.4	263.2	37.9	
6.200	9.650	0.64	0.21	0.004	47.0	0.180	0.27	0.12	0.38	0.26	23725	33375	27.8	8.0	28.0	222.2	35.9	In Cols. 11, 12 u. 13 the upper figures refer to the fire-box, the lower ones to the barrel.
1.900	11.100	0.93	0.22	0.006	43.0	0.140	0.34	0.14	0.13	0.40	26367	37467	23.4	9.8	33.3	196.7	30.0	
6.500	8,450	0.77	0.17	0.004	39.4	0.200	0.28	0.15	0.25	0.49	22558	31008	18.0	7.5	27.5	263.1	40.3	
0.430	0.615	0.70	0.18	0.007	27.8	0.370	0.44	0.17	0.15	1.37	1517	2217	10.1	4.6	14.8	150.0	67.6	
0.370	0.570	0.65	0.20	0.007	28.2	0.390	0.52	0.20	0.15	1.36	1410	2060	17.1	4.8	17.1	141.2	58.3	
ube Boilers.																		
0.643	0.643	1.00	0.15	0.020	7.7	0.220	—	—	0.09	0.33	659	1302	32.9	32.1	65.0	49.6	15.4	
1 496	0.427	0.43	0.28	0.015	32.5	0.230	—	—	0.09	0.24	13500	14990	67.5	7.4	74.9	66.7	13.3	
1.320	2.590	0.51	0.64	0.016	38.6	0.465	—	—	0.15	0.40	9169	11759	61.1	17.2	78.4	60.0	12.9	
1.500	4.150	0.43	0.29	0.010	27.4	0.217	—	—	0.17	0.43	17853	21983	89.0	20.7	109.9	95.5	15.5	
oilers.																		
0.532	0.520	1.02	—	—	25.8	0.280	—	0.77	0.10	0.25	2470	3493	—	—	—	—	—	
0.248	0.310	0.80	—	—	21.3	0.270	—	0.71	0.13	0.20	1132	1932	—	—	—	—	—	
42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60

with a wrought iron plug in which two little brass tubes are fixed, one passing nearly down to the bottom of the suspended tube, the other standing up in the cylinder. The water flows down the lower brass tube, which is the longer, to the bottom of the suspended tube, then rises up this and finally escapes as steam up the upper brass tube. Thus the whole of the suspended tubes afford effective heating surface and the arrangement is very similar to that of the FIELD tube.

Circulation.

15) The vertical cast-steel cylinder is provided with a diaphragm separating the two upper from the two lower rows of tubes. This diaphragm is dished upwards so that a channel is formed round it and this has a narrow slit all round the bottom. The feed-pipe terminates in this channel and the feed-water is thus constrained to distribute itself through the circular slit over the two lower (outside) rows of vertical tubes. It passes down through these into the annular tube at the bottom, thence through the inside vertical tubes into the bottom of the steel drum and into the hanging tubes from which it finally escapes as steam into the upper part of the top drum. Complete and uninterrupted circulation is thus established and is extremely rapid on account of the very small weight of water in proportion to the heating surface, enabling steam to be got up in a few minutes.

Steam-space.

16) The steam-space shell above the inside steel cylinder is of steel plate riveted together and has a man-hole in the crown. It contains two concentric cylinders in the central one of which the steam ascends, passes down between this and the next and does not reach the steam pipe until it arrives at the top of the space between the middle cylinder and the outside shell. This arrangement is intended to dry the steam. The water separated out flows through a pipe into a mud-tray in the lower water-space of the inside steel cylinder.

Opinion on the Ward Boiler.

17) The engineers above referred to express a very favourable opinion upon the WARD boiler on the strength of their experiments. The boiler shewed no tendency to prime at low pressures even when the combustion was forced much harder than usual. It is strong enough for much higher pressures than could be economically employed. In spite of the low funnel the draught is sufficient. Its steaming capacity is good and rapidity of evaporation excellent. The superheating in the steam space answers all requirements and the arrangement for it is so simple that it will keep efficient for a long time. The merit of the design is such that even under rough and unskilful treatment the boiler can be forced to its utmost limits both of pressure and combustion. A tube can easily be removed without disturbing any

others and therefore repairs are not difficult. The floor-area occupied is only small and the weight is very light for the high working-pressure. The treatment under steam is the same as for any other marine boiler, no specially expert stokers are required. On account of its great simplicity the life of this boiler is only limited by unavoidable gradual corrosion, and it is therefore particularly adapted for steam launches to which it is almost exclusively applied in the U. S. Navy. Recently it has been constructed of a square form in plan so as to get in more grate and heating surface and thus increase the power.

18) **IV. Rules and Regulations as to Launch Boilers.** In Germany the police regulations for boilers in general include launch boilers. In British passenger ships these boilers come under the Board of Trade Rules like main boilers. The principal classification societies, as Lloyd's, the Germanischer Lloyd, and Bureau Veritas make the same provision.

19) The table on p. p. 594 to 601 contains the principal dimensions and particulars of a number of the main, auxiliary, and launch boilers of German war-vessels. All ratios referring to *IHP* are based upon the required power for which the boilers were designed as stated in Col. 8. But in order that the corresponding ratios may be ascertained for the actual natural and forced-draught powers, the former is given in Col. 9 and the latter in Col. 10 — for every case in which they have been ascertained. Particulars of Marine Boilers.

Tenth Division.

Strength of Marine Boilers.

§ 60.

Material.

Rules.

1) The material used in the construction of marine boilers is in most countries made the subject of more or less elaborate Rules and Regulations issued by the marine authorities and the classification societies, from among which the following will be inserted in this work further on, viz. those of the German Reichsmarineamt*), the British Board of Trade**), the Germanischer Lloyd***), Lloyd's Register of Shipping †), and the Bureau Veritas ††).

Kinds of Material.

2) The principal materials used in marine boiler-making are

I. Steel, (Ingot-iron),
II. Malleable iron,
III. Copper,
IV. Brass,
V. Ordinary cast iron,
VI. Malleable cast iron.

Steel.

3) **I. Steel** is now almost exclusively used in boiler work and is usually the mild steel obtained by the Siemens Martin process. Recently however nickel-steel and so called "special" steel have been employed for the shells of exceptionally large high-pressure

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*) Vorschriften über die Prüfung und Abnahme der für den Bau von Dampfkesseln bestimmten Stahlbleche. Berlin 1892.

**) Regulations and suggestions as to the survey of the hull, equipments, and machinery of steam ships carrying passengers. London 1894.

***) Germanischer Lloyd. Reglement für die Classification und Vorschriften für den Bau und die Ausrüstung von eisernen u. s. w. Schiffen. Berlin 1896.

†) Lloyd's Register of British and Foreign Shipping. Rules for the survey and construction of engines and boilers of steam vessels. London 1894.

††) Bureau Veritas. Vorschriften für die Classification und den Bau von Schiffen. 1896.

boilers. These two materials appear to have a wide future before them. The principal parts made of steel are

a) Plates,
b) Angles, stays, and rivets,
c) Tubes.

4) a. **Rules for steel plates** usually cover **Steel plates.**

α) the checking of the thickness and weight,
β) a tensile test,
γ) a temper test,
δ) a welding test.

5) α. *The checking of the thickness and weight* is prescribed in the German Admiralty rules. **Checking the thickness and weight.** The thickness must be taken at least 40 mm from the edge and 100 mm from one corner of the plate. The following table shews the deviations from the calculated weight which will be passed, the deficiencies having the negative and the excesses the positive sign. The thickness of the thinnest part of the plate must never exceed the specified thickness. The over-weight permitted is not to be referred to every separate plate but to all the plates in each parcel which belong to one group or class for which any particular percentage of over-weight is allowed. The calculated weights are worked out at 7850 kg per cbm for steel and 7763 kg for iron. The width is to be measured square to the direction of the rolling and if the plate is not rectangular, the width of the circumscribed rectangle is to be taken as the width.

Table of Limits of Thickness and Over-weight for Boiler-plates.

Width of Plate in mm	Admissible deviation in mm from the specified thickness in mm					Admissible percentage of over-weight					Remarks
	5–6,9	7–9,9	10–14,9	15–19,9	20 and over	5–6,9	7–9,9	10–14,9	15–19,9	20 and over	
1	2	3	4	5	6	7	8	9	10	11	12
up to 1000	−0.3 +0.6	−0.3 +0.6	−0.5 +0.6	−0.5 +0.6	−0.5 +0.6	5	4	3	3	2.5	For plates below 5 mm thick the Rules of the German Iron-masters' Association apply.
1000—1300	−0.5 +0.8	−0.5 +0.8	−0.5 +0.6	−0.5 +0.6	−0.5 +0.6	5	4	3	3	2.5	
1300—1600	−0.5 +1.0	−0.5 +0.9	−0.5 +0.9	−0.5 +0.9	−0.5 +0.9	5	4	3.5	3	3	
1600—1800	−0.7 +1.4	−0.7 +1.3	−0.6 +1.2	−0.6 +1.2	−0.5 +1.0	5	4	4	3.5	3	
1800—2100	−0.9 +1.8	−0.8 +1.6	−0.7 +1.4	−0.7 +1.4	−0.6 +1.2	—	—	4	4	3	
2100—2400	—	—	−0.8 +1.6	−0.8 +1.6	−0.8 +1.6	—	—	5.5	4	4	
2400—2700	—	—	−1.0 +2.0	−1.0 +2.0	−1.0 +2.0	—	—	7	5	5	
2700 and over	—	—	−1,0 +2.5	+2.0 −1.5	−1.0 +2.5	—	—	8	6	5	

Number of tensile tests.

6. *β. The tensile test* is applied to every single plate for the German Admiralty and for this purpose samples are to be cut from two diametrically opposite corners. A strip 500 mm long and of corresponding breadth is to be prepared for the tensile test and one of 260 mm × 40 mm for the bending test. — The British Board of Trade only requires a tensile test from every fourth plate of the same thickness, but with plates above 4.5 m long a tensile test is to be taken from each end and with plates over 6 m long and 1.8 m wide as well as with those above $2\frac{1}{2}$ tons in weight, a test piece must be taken from each corner, besides sundry temper and bending tests. Lloyd's require a tensile test from every charge, but a temper test from every plate. The Veritas Rules are similar and only prescribe a tensile test of every fourth plate in the same charge, a temper test of every plate, and finally a tensile test of every shell plate. The Germanischer Lloyd leaves to the surveyors the selection of the plates to be tested.

Preparation of the test-pieces.

7) For the German Admiralty the test-pieces are heated, straightened, and annealed, the British Board of Trade prefers them not to be annealed after they are cut off the plates, the Germanischer Lloyd also only directs that the pieces are to be heated and straightened, or the entire plate annealed, the other societies do not mention this point. In all cases the test-pieces are so prepared that the skin left by the rolls on each side of the plate is not injured. The sides of the sample are to be so cut down by planing, slotting, or filing that all trace of the shearing or punching is removed the edges and slightly rounded. For the German Admiralty the extension in a length of 200 mm is measured, the same as for the three above-named classification societies, the Board of Trade takes the extension in a length of 250 mm. The German Admiralty require the samples to be 50 mm broad for plates up to 15 mm thick and for thicker plates the section tested is to be about 750 sq. mm provided that the breadth is never less than the thickness of the plate. The Germanischer Lloyd requires the section of the test-pieces to be not less than 300 nor more than 600 sq. mm. The other societies prescribe no particular sectional area.

Tensile strength.

8) The temperature in the testing house is not to be below 12° C and the tensile strength of the plates per sq. mm must be within the limits given in the following table which also shews the required extension. The Veritas Rules limit the range between the highest and lowest tensile strength of boiler material to 5 kg for steel up to 40 kg and 6 kg for steel of over 40 kg tensile

Table of the Limits of Strength for Steel Plates.

Rules of	Plates exposed to fire: Tensile strength Lowest kg	Tensile strength Highest kg	Extension %	Quality Number	Plates not exposed to fire: Tensile strength Lowest kg	Tensile strength Highest kg	Extension %	Quality Number	Remarks
German Admiralty	36	41	25	64	41	47	20	64	The Quality number, i. e. the sum of tensile strength and percentage of extension must always be at least 64.
Board of Trade	41	47	25*)	—	42.5	50	25*)	—	*) 25 % extension applies to the test-pieces 250 mm long
Lloyd's Register	41	47	20	—	41	47	20	—	Steel of lower tensile strength than 41, if otherwise of satisfactory quality can be used instead of wrought iron.
Bureau Veritas: Tensile strength kg.	34	36	38	40	42	44	46	48	The figures apply to all plates, it is recommended that soft material be used for plates exposed to fire.
Bureau Veritas: Extension %	32	29,5	27	25	23	22	21	20	

Germanischer Lloyd															Remarks
Tensile strength kg.	35	36	37	38	39	40	41	42	43	44	45	46	47	48	It is recommended that a tensile strength of 42 with corresponding extension be not exceeded for plates exposed to fire.
Extension %	26	25.5	25	24.5	24	23.5	23	22.5	22	21.5	21	20.5	20	20	

strength. Nickel-steel has 60 to 65 kg tensile strength with 20 % extension, "special" steel 54 to 60 kg with 24 % extension.

9) *γ. The temper test* is applied to a strip from every plate for the German Admiralty, from a few different plates for the Board of Trade, and from one plate out of every charge for Veritas. The Germanischer Lloyd, Lloyd's Register Society, and the German Admiralty leave the selection of the test-piece to the surveyor. The test-piece is heated, cooled in water, and then slowly bent together, preferably with a hydraulic press. After the bending there must be no cracks visible. **Method.**

Table of Temper-tests.

Rules of	Test-piece Treatment: Heat	Treatment: Temperature of the cooling water °C.	Dimensions: Length mm	Dimensions: Breadth mm	Angle of Bending	Ratio of the radius of bending to thickness of the sample	Remarks
German Admiralty	cherry-red	28—30	260	40	180°	1	
Board of Trade	"	22	250	50	180°	3	
Germanischer Lloyd	red	28	—	25	180°	2 4	for plates exposed to fire for shell plates
Lloyd's Register	dark cherry-red	22	—	—	180°	1½	
Bureau Veritas	Dark red	28	—	50	180°	1½	

Welding test. 10) δ. *The welding test* is only applied by the German Admiralty to such plates as are to be afterwards welded, it extends to one plate in each charge. For this purpose a strip composed of two pieces welded together is shaped up as described in 7) and subjected to a tensile test under which it must shew at least 90 % of the strength required of the plate. The Board of Trade only says that a welding test must be taken of such plates as are welded. The classification societies do not insist upon a welding test.

Tests. 11) b. **The Rules for angle-bars, stays, and rivets** mostly prescribe

α) a tensile test,

β) a cold and a hot bending test,

γ) a temper test.

12) α. *The tensile test* in general requires the same strength from *angle-bars* as from plates. *Stays* which are never allowed to be welded, are required by the German Admiralty to have a tensile strength of 41 to 42 kg per sq. mm with at least 22.5 % extension, by the Board of Trade 42.5 to 50 kg with 20 % extension (in a length of 250 mm), by the Germanischer Lloyd 36 to 48 kg with a corresponding extension, and by Veritas and Lloyd's Register the same strength and extension as plates. For steel *rivets* the Board of Trade requires 41 to 42 kg tensile strength with 25 % extension (in a length of 250 mm), the Germanischer Lloyd 35 to 40 kg with corresponding extension, Veritas a strength not exceeding 41 kg.

Steel angle-bars. 13) β. *The cold and hot bending test* is only applied to angle bars and generally consists in first flattening down the sample cold until the two inner faces are in one plane. The piece is then longitudinally bent or folded S-wise in three until the parts are straight and parallel to each other and not more than three times the thickness of the bar apart. At the hot test the two limbs are bent together while red-hot until their surfaces completely touch each other, the sample is then flattened out again, cut in two, and the pieces bent double lengthwise till the orginal inner surfaces are in contact, one piece red-hot, the other cold. No cracks must appear in any case.

Bending and jumping test. 14) γ. *The bending temper test* is applied to *angle-bars* by cutting longitudinal and transverse strips out of them at least 25 mm wide, heating these to a cherry red, quenching them in water of 28° C. temperature, and then bending them round a drift of a diameter equal to three times the thickness of the bar. when they must neither break nor shew cracks. — *Stay-rods* up to 35 mm diameter are for the German Admiralty tested black, those of a larger size being turned down to this diameter

before the test which consists in cooling them at a cherry red in water of 28° C. and bending them double to a radius of half the diameter of the specimen. By the Veritas Rules bars intended for steam-space and combustion-chamber stays are submitted to the same tests as plates. Steel rivet rods are not submitted to a temper test, but pieces 2 diameters long are jumped cold to half their length. Other pieces of the same size are jumped at a dull red to $^1/_3$ or $^1/_4$ of their length and afterwards punched. No cracks must appear.

15) c. **Rules for Fire and Water Tubes** embrace Steel Tubes.

α) a pressure and tensile test,

β) a cold working test,

γ) a pickling test.

16) *α. The pressure-test* consists in submitting the tubes to a cold water pressure of 40 atmos. For iron tubes up to 9 atmos. working pressure 20 atmos. are considered sufficient. Steel tubes must have a tensile strength with and across the fibre of 30 to 40 kg per sq. mm with 22 % extension in 200 mm. For iron tubes the required strength is at least 29 kg across and 35 kg with the fibre and 4 % and 6 % extension respectively. Pressure and tensile test.

17) *β. The cold working test.* The tube is first expanded by a tapered drift till its diameter is increased by 10 %. Next a ring 50 mm deep is cut off and slit open at some distance from the weld, flattened out and bent backwards into the circle again. A similar ring is flattened out and bent double through the weld. Sometimes a tube end is flanged out square 8 mm wide. Cold Working test.

18) *γ. The pickling test* consists in placing the tubes for from 12 to 14 hours in a bath of 1 volume of hydrochloric acid to 9 volumes of water, afterwards for half an hour in a bath of lime-water, and finally washing them with hot water before coating them with black varnish. After the pickling the surface of the tubes must shew no pits $^1/_2$ mm in depth or of less depth with cracked or porous bottom. Pickling Test.

19) **II. Wrought Iron** is now of so little importance in marine boiler-making that only some of the principal rules applying to it will be quoted here.*) For plates the tests include checking the thickness and weight as well as tensile, bending, and forge-tests for all material. Wrought Iron.

20) a. **Rules for Iron Plates.** It is usual to distinguish three qualities of iron boiler-plates, viz. furnace-plates, flanging plates, and shell-plates. All plates exposed to the first radiant heat of the fire must be of furnace quality. Flanging plates are used for Various plates.

*) Vorschriften für Lieferungen von Eisen und Stahl, aufgestellt vom Verein Deutscher Eisenhüttenleute. Düsseldorf 1893.

parts requiring to be flanged or dished, as end-plates, domes, necks, &c. All the rest can be of shell quality.

Tensile Test. 21) Plates up to 25 mm thickness must have the following minimum tensile strengths. A specimen will be passed if one of

Table of the Limits of Strength for Iron Plates up to 25 mm thickness.

Quality	Furnace		Flanging		Shell	
Direction with regard to fibre	parallel	across	parallel	across	parallel	across
Strength in kg per sq. mm	36	34	35	33	33	30
Extension %	18	12	12	8	7	5

its test figures fall below the prescribed minimum by 1 provided the other exceed the minimum by 1. For plates above 25 mm thick the required strength is reduced 0.5 kg for every extra 2 mm in thickness, as shewn below. Thus for instance if a

Quality	Furnace		Flanging		Shell	
Thickness mm	parallel kg	across kg	parallel kg	across kg	parallel kg	across kg
26 to 28	35.5	33.5	34.5	32.5	32.5	29.5
28 „ 30	35.0	33.0	34.0	32.0	32.0	29.0
30 „ 32	34.5	32.5	33.5	31.5	31.5	28.5
32 „ 33	34.0	32.0	33.0	31.0	31.0	28.0

shell-plate 30 mm thick is for some purpose wanted of 33 kg tensile test parallel to and 30 across the fibre, it must be ordered of flanging quality. The tensile strength must in no case exceed 40 kg.

Bending Test. 22) Strips of plate must stand bending to the angles stated below.

Cold.

Thickness	Furnace		Flanging		Shell	
	parallel	across	parallel	across	parallel	across
6 to 8 mm	130°	110°	110°	90°	70°	45
8 „ 10 „	120	100	100	80	65	40
10 „ 12 „	110	90	90	70	60	35
12 „ 14 „	100	80	80	60	55	30
14 „ 16 „	90	70	70	50	50	25
16 „ 18 „	80	60	60	40	45	20
18 „ 20 „	70	50	55	30	40	17
20 „ 22 „	60	40	50	25	35	15
22 „ 24 „	55	30	45	20	30	12
24 „ 26 „	50	20	40	15	25	10

Hot.

Any thickness	180°	180°	180°	150°	150°	100°

Forge-Test. 23) Strips about 100 mm wide must be capable of being drawn at a red heat to at least one and a half times their original width

square to the direction of rolling without shewing any cracks either on the surface or the edges. They are to stand punching at a dull red at a distance from the edge equal to half their thickness without tearing through to the edge.

24) b. **Rules for angle-bars.** For the German Admiralty angle-iron must have a tensile strength of 35 kg per sq. mm with an extension of 12 % under the same conditions as to a difference of 1 in the figures as are given in 21). In the forge-test the sample is flattened out at a bright red until the inner surfaces of the limbs are in a plane; they are then bent together either inwards or outwards, or the bar is bent longitudinally to an angle of 45°. No cracks or loose structure of the iron must become evident and the samples must bear punching without cracking. Tensile and Forge-test.

25) *Rules for rivet-iron.* The German Admiralty requires iron rivets to be put to a shearing (as well as a tensile) test and they must not break under a load of less than 28 kg per sq. mm of the total area in shear. In the forge-test they must stand bending double cold without breaking, the rivet-head must be capable of being hammered out hot till its thickness equals $^1/_5$ of the diameter of the rivet, and the stem is to be heated, flattened out, and a hole about equal to its diameter punched in it. Finally a piece of the stem two diameters long is to jumped to $^1/_3$ of its original length. No cracks must be developed in any of these tests. Shearing and Forge-test.

26) **III. Copper** is a very homogeneous metal, a very good conductor of heat, and resists all the injurious influences of bad water and acid gases of combustion. Were it not for its high cost and low tensile strength only 25 kg per sq. mm, — it would certainly be more applied in marine boiler-making. It is chiefly used for the fire-boxes of torpedo-boat boilers. (See § 56, 6.) Copper.

27) **IV. Brass,** is used on account of its high conductivity but its low strength limits its employment to tubes in boilers of low pressure. By § 1 of the German law relating to boilers the diameter of these tubes must not exceed 10 cm. Brass.

28) **V. Ordinary Cast Iron** is such an untrustworthy material for boiler-work that by § 1 of the statute above referred to its use is forbidden in any parts directly exposed to the fire, the internal diameter of which exceeds 25 cm if cylindrical and 30 cm if spherical. It was formerly employed for furnace fittings, man-hole and mud-hole doors, &c. but even for these it is continually being more and more displaced by wrought iron or steel. Ordinary Cast Iron.

29) **VI. Malleable Cast Iron** is used for the tube-doors of water-tube boilers. But as no one has hitherto succeeded in producing it of a tensile strength up to 36 kg per sq. mm it must also be regarded as an unsafe material for boiler work. Malleable Cast Iron.

§ 61.

Thicknesses.

Formulæ.

1) The thickness of boiler-plates, the material being assumed to be uniform, depends in the first place upon the working pressure, next upon the form of the part for which a plate is used and the staying of it, and finally upon the degree of wear to which it is exposed from contact with either combustion-gases, water, or steam. The strengths of marine boilers are calculated from the following formulæ the constants in which are based upon the practical experience of many years and take account of the various stresses as well of the gradual deterioration.

Subdivisions.

2) The plates of a boiler may be classified as

I. Flat Plates, viz.
   a) End-plates,
   b) Combustion-chamber plates,
   c) Tube-plates, and

II. Cylindrical Plates, viz.
   d) Shell,
   e) Furnaces.

Definitions.

3) The following formulæ are all on the metrical system, to which the English formulæ have been converted.

The definitions are

δ the thickness in cm,

p the working pressure in atmospheres,

l the pitch of longitudinal and combustion-chamber stays in cm,

l_1 the pitch of stays in one row in cm,

l_2 the distance apart of two rows of stays in cm,

F the stayed area in sq. cm,

T the tensile strength of the material of the plate in kg per sq. cm (for Veritas in sq. mm),

$T_{\prime\prime}$ the shearing strength of the material of the rivets in kg per sq. cm,

T_b admissible bending stress for the material of the plate in kg per sq. cm,

W the extreme length of combustion-chamber measured from back of tube-plate to front of combustion-chamber back in cm,

D the extreme internal diameter of the boiler in cm,

D_1 the extreme external diameter of a furnace in cm,

L the length of a furnace, or the distance between stiffening rings where they are fitted, in cm,

E the least horizontal pitch of tubes in cm,

d_r the diameter of the rivet circle of stay-washers in cm.

d_f the internal diameter of tubes in cm,
d_n the diameter of rivets in cm (for Veritas in mm),
d_s the diameter of longitudinal and combustion-chamber stays in cm,
e the pitch of rivets in cm,
a the number of rivet-sections in one pitch,
φ_k the factor of safety of the plate,
φ_n the factor of safety of the riveted seam,
ϱ_k the percentage or ratio of strength of the plate in the seam to that of the full plate,
ϱ_n the percentage or ratio of strength of the rivets to that of the full plate.

4) **I. Flat Plates.** a. **End-plates.** The thickness of flat plates depends upon the extent of the loaded and unsupported area, the method of staying adopted, and the working pressure. These considerations are embodied in the following formulæ, those of Lloyd's Register and the Germanischer Lloyd being identical. Their constants for the various cases differ somewhat however. **Formulæ for End-plates.**

$$\left.\begin{aligned} &\text{Board of Trade, } \delta = \sqrt{\frac{p(F-38)}{C}} - 0.158 \text{ cm} \\ &\text{Lloyd's Register, } \delta = l\sqrt{\frac{p}{C}} \text{ cm} \\ &\text{Germanischer Lloyd, } \delta = Cl\sqrt{p} \text{ cm} \\ &\text{Bureau Veritas, } \delta = 1.5 + \frac{Cp}{T}\sqrt{l_1^2+l_2^2} \text{ mm} \end{aligned}\right\} \quad . \; . \quad (292)$$

The constant C has the following values;

5) *a. By the Board of Trade Rules*, for *iron* end-plates not in contact with fire, **Board of Trade Constants for End-plates.**

2880 for nutted stays and riveted doubling strips fitted in way of the stays, equal to the end-plate in thickness and $^2/_8$ the pitch of the stays in width;

if the end-plate is doubled all over, the Board of Trade allows a still higher constant;

2700 if instead of the doubling strips, a riveted washer is fitted to each stay of the same thickness as the end-plate and at least $^2/_3$ of the pitch of stays in diameter;

1800 for nutted stays with washers of $^2/_8$ the thickness of the end-plate and 3 times the diameter of the stays;

1620 for nutted stays without washers;

for iron end-plates in contact with steam on one side and combustion gases on the other,

1080 when the staying is arranged as for the above constant of 1800;

972 when the staying is arranged as for the above constant of 1620;
648 when the stays are screwed into the plates and riveted over;
for iron end-plates in contact with flame on one side and water on the other,
1440 for stays screwed into the plates and nutted;
1080 „ „ „ „ „ „ „ riveted over.
For *steel plates* the above constants may be increased by 10% for stays screwed in and riveted, and by 25% when the stays are fastened on a better system.

Lloyd's constants for Flat-plates.

6) *β. By the Rules of Lloyd's Register C* is

4320 for steel and 3420 for iron plates and stays with double nuts, the plates having doubling strips of their own thickness and at least $^2/_3$ of the pitch of stays in one row in width riveted on in way of the stays;

3960 for steel and 3150 for iron plates as above when the width of the doubling strip is only $^2/_3$ of the distance apart of the rows of stays or when each stay has a riveted washer of the thickness of the end-plate and at least $^2/_3$ of the pitch of stays in diameter;

3600 for steel and 2880 for iron plates as above when the washers are $^1/_2$ the thickness of the plate and only $^2/_3$ of the pitch of stays in diameter;

3330 for steel and 2700 for iron plates as above when the washers are $^1/_2$ the thickness of the plate and $^1/_3$ of the pitch in diameter;

3150 for steel and 2520 for iron plates when the stays have only double nuts with no washers;

2430 for steel plates of 14 mm thickness and above with nutted stays;

2160 for steel plates from 11 to 14 mm and iron plates above 11 mm thick with screwed and nutted stays;

1980 for steel and iron plates 11 mm and less in thickness with stays and nuts;

1800 for steel and iron plates above 11 mm thick with screwed stays riveted over;

1600 for steel and iron plates 11 mm and less in thickness with riveted stays.

For end-plates in contact with steam on one side and combustion-gases on the other the above constants are to be increased by 20%.

End-plates doubled completely on the outside with a doubling plate of $^2/_3$ their thickness are calculated by the following formula, the above constants being applied,

$$\delta = \sqrt{\frac{p\,l^2}{C} - \left(\frac{\delta_1}{2}\right)^2} \text{ cm} \quad \ldots\ldots\ldots\ldots \quad (202)$$

where δ_1 is the thickness of the doubling plate.

7) γ. *By the Germanischer Lloyd's Rules*, C is for *iron* flat plates not in contact with combustion gases, **Germanischer Lloyd's Constants for flat plates.**

0.019 for nutted stays with washers of the thickness of the plate and $^4/_5$ the pitch of stays in diameter;

0.020 as above, for washers $^5/_6$ the thickness of the plate and $^3/_5$ the pitch of stays in diameter;

0.021 as above, for washers $^2/_3$ the thickness of the plate and $^2/_5$ the pitch of stays in diameter;

0.023 for screwed stays and nuts;

0.024 for stays screwed in and riveted.

For iron flat plates in contact with combustion-gases on one side and steam on the other, C is

0.025 for screwed and nutted stays;

0.027 for screwed and riveted stays;

the thickness is to be 10 % greater than that given by these constants. The protection of these plates by baffle-plates over the stay-ends in the up-take is strongly recommended.

For steel plates the thicknesses arived at as above can be reduced by 10 % for screwed and riveted stays, but by 10 to 15 % when the fastenings are of a superior character.

8) δ. *By the Bureau Veritas Rules* C is **Veritas Constants for flat plates.**

0.481 for stays with nuts and washers on both sides of the plate, the outside washer riveted to the plate, at least $^3/_4$ the thickness of the plate and 0.6 of the pitch of stays in diameter;

0.542 for screwed stays with internal and external nuts and washers at least 0.4 of the pitch of stays in diameter; the external washer is to have at least $^2/_3$ the thickness of the plate and more if its diameter exceeds $1\,^1/_2$ times the diameter of the nuts measured over the cants;

0.578 for screwed stays with external nuts, or combustion-chamber stays nutted at one end and having the other end screwed into a thick plate and riveted over;

0.735 for screwed stays riveted over.

Flat plates exposed to steam on one side and combustion-gases on the other are not required to be of increased thickness if protected by baffle plates. If not, they are to be calculated from this formula

$$\delta = 3 + \sqrt{(l_1^2 + l_2^2)\frac{p\,C}{0.9\,T}}\ \text{cm} \quad \ldots\ldots\ldots \quad (292^b)$$

where C has the above values. The tensile strength is *always* to be taken at $T = 33$ kg per sq. mm for iron but for steel at the minimum strength of the material tested according to § 60,8.

Combustion-chamber plates.

9) b. **Combustion-chamber plates** are to be calculated from the same formulæ as end-plates, the corresponding constants being used.

Tube-plates.

10) c. **For Tube-plates** the Germanischer Lloyd and Bureau Veritas give no special formula besides Eq. (292) but the Board of Trade requires that if any compressive stress is transmitted by the combustion-chamber sides to the back tube-plate it must not exceed 560 kg per sq. cm for iron and 700 for steel tube-plates. Lloyd's Rules fix this stress at 900 kg both for iron and steel. The regulations of the two Societies may therefore be embodied in one formula

$$\delta = \frac{p W E}{C(E - d_t)} \text{ cm} \quad \ldots \ldots \ldots \ldots \quad (293)$$

where C is

1220 for iron back tube-plates to Board of Trade,
1400 „ steel „ „ „ „ „ „ ,
1800 for iron and steel back tube-plates to Lloyd's Rules.

The latter Rules further direct that the thickness of steel front tube-plates between the boxes of tubes is to be calculated from Eq. (292) with $C = 2520$ and $l =$ the mean pitch of stay-tubes also that the same formula is to be applied to the unbored portion of tube plates between the boxes — in general, when l signifies the horizontal distance between two rows of tubes some of which are stay-tubes and C has the following values —

2160 where in the rows having stay-tubes, every third tube is a stay-tube without an external nut;
2340 as above, but with external nuts;
2520 when in the respective rows every other tube is a stay-tube without an external nut;
2700 as the last, but with external nuts;
2880 when in the respective rows every tube is a stay without an external nut;
3060 as the last, but with external nuts.

Bach's Formulæ.

11) BACH'S latest investigations into the admissible load on stayed flat plates have led to the following results;*)

for screwed stays riveted over

$$\delta = l \sqrt{\frac{p}{3.6\left(1 - 0.7\,\frac{d_s}{l}\right) T_b}} \text{ cm,}$$

for stays with double nuts and washers

$$\delta = l \sqrt{\frac{p}{3.6\left(\dfrac{1 - 0.7\,\dfrac{d_s}{e}}{1 - 1.8\,\dfrac{d_s}{l}}\right) T_b}} \text{ cm,}$$

*) Zeitschrift des Vereines deutscher Ingenieure. Berlin 1894. p. 341.

for stays with riveted washers about equal in thickness to the plates

$$\delta = l\sqrt{\frac{p}{4\left(\frac{1}{1-0.9\frac{d_s}{l}}\right)T_b}}\ \text{cm.}$$

BACH recommends 600 kg per sq. cm as an admissible bending stress for tough mild steel and wrought iron, but considers it safe to go as high as 800 if necessary.

12) **II. Cylindrical Shells.** d. **The Shell** of a boiler depends for its strength upon that of the longitudinal seams, its thickness δ therefore increases with the diameter D, the working pressure p, and the factor of safety φ, but may decrease with the strengths T of the material and ϱ of the seams. This principle is the foundation of the various classification Societies' formulæ for the thickness of cylindrical boiler shells. That of Lloyd's Register and the Bureau Veritas unite the factor of safety φ and the strength of the material in one constant C. The formulæ follow below. Formulæ.

$$\left.\begin{array}{ll}
\text{Board of Trade and Germanischer Lloyd,} & \delta = 0.5\dfrac{\varphi_b\, p\, D}{\varrho\, T}\ \text{cm} \\[2ex]
\text{Lloyd's Register for iron boilers, ...} & \delta = \dfrac{p\, D}{\varrho\, C}\ \text{cm} \\[2ex]
\text{,, \quad ,, \quad ,, steel \quad ,,} & \delta = \dfrac{p\, D}{\varrho\, C} + 0.317\ \text{cm} \\[2ex]
\text{Bureau Veritas, ...} & \delta = 0.05\dfrac{p\, D}{\varrho\, C} + 1\ \text{mm}
\end{array}\right\}\ . \quad (294)$$

These formulæ have reference to the following rules;

α) for the factor of safety of the plate φ_k,

β) for the percentage of strength in the seam ϱ,

γ) for the tensile strength T,

δ) for the factor C.

13) α. *The Factor of safety* of the plate φ_k is found by the Board of Trade Rule as follows. Factor of safety of the Plate.

For cylindrical boilers of the best iron, all holes drilled in place, double butt-straps at least $^5/_8$ of the thickness of the plate, all riveting at least double with a percentage of strength of the full plate of at least 75, and the whole of the work carried out under the inspection of the surveyor, the factor of safety φ_k, can be taken at 5. For exactly similar boilers of steel it is 4.5. If the above conditions are not complied with, the factor 5 or 4.5 is to be increased by the fraction given for each particular case in the second column of the following table:

Table of Constants to be added to the Factor of Safety φ_k of the Plate for cylindrical Shells.

Reference Letter	Constant	Conditions under which the Constant is to be added.
A*)	0.15	When the holes in the longitudinal seams fit accurately but are not drilled together in place *after* the plates are bent,
B**)	0.3	As A, but holes not drilled together in place *before* the plates are bent,
C	0.3	As A, but holes punched instead of being drilled *after* plates are bent,
D	0.5	As A, but holes punched instead of being drilled *before* plates are bent.
E*)	0.75	When the holes in the longitudinal seams do not fit accurately,
F	0.1	When the holes in the circumferential seams fit accurately but are not drilled together in place *after* the plates are bent,
G**)	0.15	As F, but holes drilled *before* the plates are bent,
H	0.15	As F, but holes punched *after* the plates are bent,
I**)	0.2	As F, but holes punched *before* the plates are bent,
J*)	0.2	When the holes in the circumferential seams do not fit accurately,
K	0.2	When the longitudinal seams are lapped and double-riveted instead of having double butt-straps,
L	0.1	As K, but with treble-riveted lap,
M	0.3	When the longitudinal seams have only single butt-straps double-riveted,
N	0.15	As M, but treble-riveted,
O	1.0	When any of the longitudinal seams are single-riveted,
P	0.1	When the circumferential seams have single butt-straps double-riveted.
Q	0.2	As P, but single-riveted,
R	0.1	When the circumferential seams have single-riveted double butt-straps.
S***)	0.1	When the circumferential seams are lapped and double-riveted,
T	0.2	As S, but single-riveted,
U	0.25	When the circumferential seams are lapped and the plates do not fit accurately,
V***)	0.3	When the boiler is double-ended or of unusual length and the circumferential seams arranged as P, R and S, but if Q and T apply V is to be taken as = 0.4 instead V = 0.3,
W*)	0.4	When the longitudinal seams are not well shifted clear of each other.
X*)	0.4	When the quality of the iron is doubtful or the Surveyor is not satisfied that it is the best,
Y	1.65	When the boiler has not been accessible to the Surveyor during the whole period of construction.

By the Rules of the Germanischer Lloyd

for a thickness of 10 mm and under 15 mm: $\varphi_k = 5.00$

" 15 " " " 20 " : $\varphi_k = 4.85$

" 20 " " " 25 " : $\varphi_k = 4.80$

*) In these cases the constant may be increased when the material and workmanship are very doubtful or very defective.

**) In these cases the constant can be either reduced or neglected accordingly as the holes are either rimered or re-drilled in place.

***) In these cases S = 0.1 and V = 0.3 can be neglected when the middle circumferential seam has double-riveted double butt-straps or treble-riveted single butt-straps with a percentage of strength of not less than 65% of the full plate. The circumferential seams at ends must be at least double-riveted.

for a thickness of 25 mm and under 30 mm: $\varphi_k = 4.75$
„ 30 „ „ „ 35 „ : $\varphi_k = 4.70$
„ 35 „ „ „ 40 „ : $\varphi_k = 4.65$
„ 40 „ „ above : $\varphi_k = 4.60$

14) *β. The ratio ϱ of the strength of the riveted seam* to that of the full plate is expressed by the Board of Trade and Lloyd's as a percentage, by the Germanischer Lloyd and Veritas as a fraction. Strength of the seam.

The Board of Trade formula for the strength of the plate in the seam expressed as a percentage of the strength of the full plate is

$$\varrho_b = 100\left(\frac{e - d_n}{e}\right)\%$$

and the percentage which the strength of the rivets is of that of the full plate . . (295)

$$\varrho_n = 100\left(\frac{\frac{\pi}{4} d_n^2 n}{e\, d}\right) \varphi_n\, \%$$

where n is the number of rivets in one pitch and φ_n the factor of safety for the rivets in shear is to be taken as

$\varphi_n = 1$ for laps,

$\varphi_n = 1.75$ for double butt-straps.

The greatest admissible pitch e is given in § 62.

Lloyd's Society has retained the former of these formulæ but restricts the application of the latter one to the case of iron rivets and iron plates with punched holes.

When they are drilled

$$\varrho_n = 90\left(\frac{\frac{\pi}{4} d_n^2 n}{e\, \delta}\right) \varphi_n\, \% \qquad \text{. . . . . (295}^a\text{)}$$

For steel boilers the factor 90 is to be replaced by

85 when the rivets are also of steel,

70 when the rivets are of iron.

The values of φ_n remain the same.

The Germanischer Lloyd and Bureau Veritas do not calculate with a percentage but merely with the direct ratio of strength in the seam to that in the full plate, so that

$$\varrho_b = \frac{e - d_n}{e}$$

Both these societies further require the shearing strength of the rivets in the longitudinal seam to be at least equal to the tensile strength of the plate as weakened by the rivet-holes, so that by § 62, 12 we must have the relation

$$n\frac{\pi}{4}d_n^2 = c\,\delta$$

and substituting in this the corresponding value for δ from Eq. (294),

$$n\frac{\pi}{4}d_n^2 = \frac{\varrho_n p D}{2T_n}c$$

$$\varrho_n = \frac{n\cdot\frac{\pi}{4}d_n^2 \cdot 2T_n}{p D c}$$

$$\varrho_n = \frac{1.57\,n\,d_n^2\,T_n}{d D c}$$

T_n is here equal to the minimum shearing strength of the rivets as determined by tests, or if none have been taken T_n is to be 0.875 T for iron rivets and 0.8 T for steel rivets. The Board of Trade and Germanischer Lloyd direct that the smaller value of ϱ_b and ϱ_n got from the above formulæ is to be substituted for ϱ in Eq. (294). By Lloyd's Rules a higher value of ϱ than results from calculating ϱ_b or ϱ_n cannot be used in Eq. (294) unless established by actual tensile tests.

Tensile Strength. 15) γ. *The tensile strength* T is put by the Board of Trade at $T =$ 3300 kg per sq. cm for iron and = 4300 kg per sq. cm for steel, rising according to the actual tensile strength to a maximum of 5000 kg per sq. cm provided the surveyor has witnessed all the tests. The Germanischer Lloyd makes T = the minimum tensile strength as obtained from the tests of the material.

Constant C. 16) δ. *The Constant* C for Lloyd's formulæ in Eq. (294) is taken for *iron boilers* from the following Table

Description of longitudinal seam	Plates up to 13 mm thick	Plates from 13 to 19 mm thick	Plates over 19 mm thick
Lap joint, holes punched	10.898	11.600	11.953
„ „ drilled	11.953	12.650	13.359
Double butt-strap, holes punched	11.953	12.650	13.359
„ „ „ , „ drilled	12.650	13.359	14.002

For *steel boilers* the values of C in the latter of Lloyd's two formulæ in Eq. (294) are

$C = 20.811$ when the longitudinal seams are lapped,

$C = 21.655$ when the longitudinal seams have double butt-straps of unequal width,

$C = 22.499$ when the longitudinal seams have double butt-straps of equal width.

The internal buttstrap must have at least $^3/_4$ the thickness of the shell plate. In the Veritas Rules $C =$ the minimum tensile strength of the material T divided by 5.5 for steel and 4.4 for iron boilers.

17) e. **For Furnaces** of plain cylindrical form the thickness is in general determined by the formula General Formulæ.

$$\delta = \sqrt{\frac{L\,D_1\,p}{C}}\ \text{cm} \quad \ldots\ldots\ldots\ldots \quad (296)$$

The Board of Trade for greater safety insert $L + 30.5$ instead of L, so that the formula becomes

$$\delta = \sqrt{\frac{(L + 30.5)\,D_1\,p}{C}}\ \text{cm} \quad \ldots\ldots\ldots \quad (296^a)$$

The Germanischer Lloyd puts the constant C outside the radix, thus

$$\delta = C\sqrt{L\,D_1\,p}\ \text{cm} \quad \ldots\ldots\ldots\ldots \quad (296^b)$$

18) The constant C in Eq. (296) is Constant C for Eq. (296).
in Lloyd's Rules
$C = 75620$ for *iron* plates not exceeding 14 mm in thickness (for thicker iron plates see 23),
in the Veritas Rules
$C = 58800$ for *iron* plates, when the furnace is perfectly circular and the longitudinal seam is either welded or butted with a strip, or if lapped when the lap is bent or dished so as to preserve the perfectly circular form and double-riveted,
$C = 50400$ for *iron* plates when the furnace is not perfectly circular or the seam merely lapped.

These constants may be multiplied by 1.2 for plates of soft steel or best quality iron with a tensile strength of 36 kg per sq. mm and 16 % extension with and 34 kg and 10 % extension across the fibre.

19) By the Board of Trade Rules the constant C in Eq. (296^a) for *iron* plates is to be taken from the following table in each particular case. Constant C for Eq. (296^a).

Table of Board of Trade Constants.

Butted furnaces.

Character of Seam	Drilled Holes	Punched Holes
Longitudinal seams welded	75960	
„ „ double-riveted single strip . . .	75960	71740
„ „ single „ „ „ . . .	67520	63300
„ „ „ „ double „ . . .	75960	71740

Lapped Furnaces.

Character of Seam	Drilled Holes	Punched Holes
Longitudinal seams double-riveted, furnaces circular .	67520	63300
„ „ „ „ „ not „ .	63300	59080
„ „ single „ „ circular .	59080	54860
„ „ „ „ „ not „ .	54860	50640

For the vertical fire-boxes of auxiliary and similar boilers the above constants are to be reduced by 10%. If either the workmanship or the material is not of the best quality, only 67520 is to be put for 75960, 59080 for 67520, and 50640 for 50080. If neither workmanship nor material is first-class the constant C is still further diminished according to circumstances and the judgment of the surveyor. A condition of good workmanship is that the longitudinal seams are to have either double-riveted single strips or single-riveted double strips, the holes to be drilled in place after the plates are bent, the plates to be afterwards taken apart, the burr removed from the holes and the latter slightly countersunk on the outside.

Constant C for Eq. (296b).

20) The Germanischer Lloyd puts the constant C in Eq. (296b) at

$$C = 0.00385$$

Least thickness of Furnaces.

21) Limiting values for the thickness of furnaces to secure them against collapse are prescribed by all the classification societies. The least admissible thickness is calculated from the expression.

$$\delta = \frac{p D_1}{C} \text{ cm} \quad \ldots\ldots\ldots\ldots \quad (297)$$

The constant C is

Board of Trade Constant C.

22) α. for the Board of Trade,

632 for plain cylindrical furnaces of iron, the corresponding compressive stress therefore does not exceed 316 kg per sq. cm,

949 for MORISON'S patent furnaces of steel,

984 for corrugated or ribbed furnaces (FOX'S and BROWN'S patents) of steel (see Pl. 41);

Lloyd's Constant C.

23) β. for Lloyd's

562 for plain cylindrical furnaces under 14 mm in thickness.

618 for plain cylindrical steel furnaces over 14 mm in thickness,

731 for furnaces with an ADAMSON ring at about the centre,

801 for furnaces with 2 ADAMSON rings;

Veritas Constant C.

24) γ. for Veritas

560 for plain cylindrical iron furnaces,

630 for plain cylindrical steel furnaces.

The Bureau Veritas recommends that plain cylindrical furnaces should be divided transversely into portions of such length that the thickness does not require to exceed 1.6 cm.

25) In order to take wear and tear into account Eq. (297) has been modified as follows — Thickness of Furnaces with regard to wear and tear.

$$\delta = \frac{p D_1}{C_1} + 0.3 \text{ cm} \quad \ldots \ldots \ldots \ldots \quad (297^a)$$

The various values of C_1 in this expression are

26) *α*. for Lloyd's, — Lloyd's Constant C_1.

1036 for spirally corrugated furnaces,

1063 for HOLMES'S patent furnaces,

1125 for furnaces with 3 ADAMSON rings and for corrugated furnaces where the corrugations are 3.8 cm deep and of 15 cm pitch, the tensile strength being not under 41 kg per sq. cm,

1304 for ribbed furnaces, the ribs being pitched not more than 23 cm apart,

1410 for FOX'S or MORISON'S patent furnaces of a tensile strength between 41 and 47 kg per sq. mm;

27) *β*. for the Germanischer Lloyd, — The Germanischer Lloyd Constant C_1.

740 for plain cylindrical furnaces without ADAMSON rings,

900 for plain cylindrical furnaces with 1 ADAMSON ring when the unsupported length between the ring and either end does not exceed 122 cm,

1010 for plain cylindrical furnaces with 2 ADAMSON rings when the unsupported length between the stiffeners does not exceed 79 cm,

1125 for plain cylindrical furnaces with 3 ADAMSON rings when the unsupported length between the stiffeners does not exceed 61 cm,

1220 for plain cylindrical furnaces with 4 ADAMSON rings when the unsupported length between the stiffeners does not exceed 50 cm, also for FOX'S or MORISON'S patent furnaces of a tensile strength between 35 and 41 kg per sq. cm, the thickness being not less than 0.8 cm, the depth of corrugations at least 3.8 cm and the plain part at the mouth not more than 25.4 cm long; also for PURVES'S patent furnaces with ribs projecting at least 3.5 cm above the plain part, internal grooves not exceeding 1.9 cm in depth, the pitch of the ribs not more than 22.9 cm, and the plain end not exceeding 15.2 cm in length,

1025 for HOLMES'S patent furnaces;

The Bureau Veritas Constant C_1.

28) γ. For the Bureau Veritas,

1390 for corrugated furnaces where the corrugations are at least 4 cm high and of 15 cm pitch,

1300 for ribbed furnaces where the ribs are at least 3.5 cm high and of 23 cm pitch; the material of both these descriptions of furnaces must have a tensile strength between 41 and 47 kg per sq. mm.

Bach's formula.

29) BACH*) has propounded the following formula for the thickness of circular *plain* furnaces based upon FAIRBAIRN'S experiments of 1858 and GRASHOF and LOVE'S commentaries upon them, also upon the accounts of RICHARDS'S investigations published in 1881, but especially upon the tests of 18 different plain cylindrical and corrugated furnaces made at the Imperial Dockyard at Danzig between 1887 and 1892.

$$\delta = \frac{p D_1}{2000}\left(1 + \sqrt{1 + \frac{a}{p}\,\frac{L}{L + D_1}}\right) \text{ cm} \quad \ldots\ldots \quad (298)$$

Values of the Constant a.

30) The constant a of Eq. (298) depends upon the degree of perfection of the circularity of the furnace. Assuming absolute circularity, if it were possible to attain it, and that no other forces were acting upon the furnace but the uniform radial fluid pressure, a would be zero. In reality however horizontal furnaces are exposed to excessive heat on one side, giving rise, especially with deposits on the crown from impure feed-water, not only to serious racking and deformation but occasionally to collapse. The value of a is intended to take these circumstances into account, so that it can be much lower for the vertical fire-boxes of auxiliary boilers which are not subjected, or at any rate to a far smaller extent than horizontal ones, to one-sided heating. BACH accordingly puts, for horizontal furnaces

$a = 100$ when the longitudinal seams are lapped and double-riveted, the plates 12 to 13 mm thick, and the circular form carefully adhered to.

$a = 80$ when the longitudinal seams are either welded or fitted with double strips and the circular form well preserved; where such furnaces are very short, the figure 80 may be reduced;

for vertical furnaces

$a = 70$ for longitudinal seam lapped and circular form,

$a = 50$ „ „ „ welded „ „ „ .

The above values for a have been adopted by the International Association of Boiler Inspection Societies in its "Hamburger Normen" (Hamburg Rules).

*) C. BACH. Die Maschinen-Elemente. Edition VI, Stuttgart 1897. p. 174

31) For low pressures and especially for land boilers exposed to hard wear the thicknesses calculated from BACH'S formula are to be increased as below Limits of Thickness for Furnaces.

for $p = 5$ atmos. and under, by 1.5 mm,
„ $p = 6$ „ „ „ „ 1.0 mm,
„ $p = 7$ „ „ „ „ 0.3 mm.

This augment is not applied with higher pressures, because a proportionately liberal allowance of strength becomes impracticable. The lowest limit of thickness is $\delta = 0.7$ cm whatever the result of the calculation may be. The Germanischer Lloyd does not go below 0.8 cm for corrugated furnaces. The Bureau Veritas recommends (see 24) 1.6 cm as the greatest thickness for plain furnaces, but in England as much as 1.9 cm has been adopted for some years (KILVINGTON and TAYLOR*) and BACH**) quotes examples of land boilers in Württemberg with furnaces up to 175 cm diameter and 2.3 cm thick, the conductivity of which leaves nothing to be desired. He therefore remarks that "*the thickness of furnace-plates has not nearly so great an influence upon their conductivity as was formerly assumed*".

§ 62.

Riveting.

1) **I. Carrying out the work.** Boiler plates are united by riveting. In modern boiler-work the largest possible plates are adopted in order to reduce the number of riveted joints, — seams and butts, to the fewest. The holes may be either punched or drilled. The former process was usual with low-pressure boilers as it was cheaper than drilling. The steel plates now used for boiler work must be annealed after punching to relieve the stresses it sets up in the plates and the holes must afterwards be rimered to the proper diameter. Punching therefore becomes almost as costly as drilling and consequently the rivet holes of high-pressure boilers are almost always drilled. It is best to drill the holes after the plates are bent and assembled exactly as they are intended to be in the finished boiler, thus insuring an accurate fit. Drilling the plates.

2) After drilling, the plates are taken apart and the holes cleared of burr on both sides by countersinking. Before they are again bolted up for riveting all the jointing surfaces should be care- Cleaning up the joint surfaces.

*) Transactions of the North-East Coast Institution of Engineers and Shipbuilders.

**) Zeitschrift des Vereines deutscher Ingenieure. 1891. p. 93.

fully freed from rust, dirt, &c. It is to be strongly recommended that circular furnaces should have their ends cleaned up in the lathe where they fit into the front plate and tube plate and that the jointing surfaces of these plates where flanged to receive the furnace ends should be bored out clean to fit them. In this way permanently tight riveting is to be secured.

Hand and machine riveting.

3) The rivets may be closed either by hand or the hydraulic riveter. In modern boiler-work machine riveting is applied wherever it is possible. The entire rivet, not merely the point, must be slowly heated for from 15 to 20 minutes in a special stove. Care must also be taken that the machine does not release the point until the red heat has gone out of it, thus increasing the friction between the surfaces and therefore the strength of the joint.

Caulking.

4) Riveted joints must be caulked on account of the inequalities of the plates. The friction is again very considerably augmented by the caulking, so that for high pressures it is advisable to caulk all lap joints inside and out. In hand-riveting the heads are also caulked which is not considered necessary in hydraulic riveting although it would be an improvement for the sake of the increased friction.

Frictional resistance.

5) Between 1892 and '94 BACH*) made a series of searching experiments at Stuttgart on about 350 specimens and found that the frictional resistance of well-designed and executed single-riveted lap-joints is from 1000 to 1800 kg per sq. cm of rivet section which is quite sufficient by itself to carry any stress that ought to come upon the joint.

Arrangement of rivets.

6) The rivets in a seam may be disposed in one, two, three, or four rows, as required. When the rivets are placed in parallel rows and exactly behind each other in a direction square to the seam the arrangement is called *chain-riveting* (Pl. 40, Figs. 1 & 4), but if the rivets in one row are mid-way between those in the next row the riveting is called *zigzag* (Pl. 40, Figs. 2 & 5). In the latter arrangement the several rows of rivets can be kept closer together, insuring tighter contact of the intervening surfaces, so that less material is required and the boiler comes out lighter than if chain-riveted throughout.

Classification of joints.

7) In boiler-work two kinds of joints are to be distinguished,
the lapped joint
and the double-buttstrapped joint.

*) C. BACH. Die Maschinenelemente. Edition VI. Stuttgart 1897. p. 142.

8) *The lapped joint* is made by lapping the plates over each other at the seam and the riveting may be either single, double, or treble according to the strength required (Pl. 40, Figs. 1 to 3). The old low-pressure boilers were entirely put together with lapped joints and in modern high-pressure boilers the several courses of plating composing the shell are lapped and united by treble-riveting. The furnaces and combustion chambers are also lap-jointed. A disadvantage of the lap-joint is that the plates it unites must be drawn down to a taper at corners and that the adjacent plate must be to a certain extent bulged at these places. But the worst fault of the lap-joint is that it puts a large bending stress on the plate and must therefore be replaced in all longitudinal seams for high pressures and large diameters by the double buttstrap joint. **Lapped joint.**

9) *The double buttstrap joint* is universally adopted for the longitudinal seams of large cylindrical boiler-shells the thickness of which is so considerable that lap-joints cannot be applied. The plates are fitted close and square together at the butt and in rare cases scarphed over each other. A buttstrap is placed over the joint on both sides and connected with both plates by single, double, or treble riveting (Pl. 40, Figs. 4 to 12). The Board of Trade requires the buttstraps to be at least $^5/_8$, the Germanischer Lloyd at least $^3/_4$ the thickness of the plates they connect, but if they are terminated against the edge of the adjacent course of plating it is well to keep them equal in thickness to the plates. The double-buttstrap joint has not the above-named disadvantage of the lap joint, but to insure tight work it is advisable to taper the buttstrap under the adjacent outside course and to hollow the plate out to receive the tapered end of the buttstrap under it. Figs. 13 to 16, Pl. 40 shew examples of double buttstrap joints in the shells of cylindrical boilers. **Double Buttstrap joint.**

10) **II. Calculation of the strength.** The rules of the classification societies for riveted joints are based upon the strength of the materials, whereas BACH, taking the actual state of affairs into account, starts with the frictional resistance of the joint because the rivet having been put in hot no longer accurately fills the hole when cold. As a rule the following are the particulars to be determined for the riveted joints of marine boilers; **Calculation of the strength of the joint.**

a) the strength of the plate in way of the rivets in proportion to that of the full plate ϱ_b,
b) the strength of the rivets in proportion to that of the full plate ϱ_n,
c) the diameter of the rivets d_n,
d) the pitch e,

e) the distance apart of the several rows of rivets,

f) the distance of the centre of the outside row of rivets from the edge of the plate.

Strength in the plate.

11) a. **The ratio ϱ_b.** The full plate measured between the centres of two rivets has the length e and the thickness δ, consequently the sectional area is $e\,\delta$, whereas the portion of plate left between two rivet-holes is shorter by the diameter of one rivet and its sectional area is therefore $(e - d_n)\,\delta$. The ratio of the sectional areas is then expressed by

$$\varrho_b = \frac{e - d_n}{e} \quad \text{or} \quad \frac{100\,(e - d_n)}{e}\ \%$$

Strength in the rivets.

12) b. **The ratio ϱ_n.** If within one pitch e (meaning the distance from centre to centre of rivets in the outside row) there are n rivets, their total sectional area is $n\,\frac{\pi}{4}\,d^2$ and this multiplied by the factor of safety φ_n of the riveting (see Eq. 295), according to whether the joint is lapped or double-buttstrapped, gives $n\,\frac{\pi}{4}\,d^2\,\varphi_n$ as the actual section in shear which is to be compared with the section of the full plate, thus

$$\varrho_n = \frac{0.7854\,n\,\varphi_n\,d_n^2}{e\,\delta} \quad \text{or} \quad \frac{100\,(0.7854\,n\,\varphi_n\,d_n^2)}{e\,\delta}\ \%.$$

But the Board of Trade allows this percentage to be applied only to the shells of steel cylindrical boilers when the longitudinal seams have double buttstraps and iron rivets are used: if these seams are lapped the rivet section in one pitch must be at least $1\,{}^5/_8$ times as great as the plate section between the rivets. The Board also assumes the shearing strength of steel rivets to be 23 tons per sq. inch (36 kg per sq. cm) while the tensile strength of the steel plate is taken as at least 28 tons (44 kg per sq. cm), so that for steel rivets and steel plate the rivet section in one pitch must be $\frac{28}{23}$ times as great as the plate section. The factor of safety of the riveted seams of steel boilers is always taken at 4.5, that of the plate at $\varphi_k = 4.5$ plus the particular constant given in § 61, 13 which corresponds to the case. Thus for steel boilers with steel rivets

$$\varrho_n = \frac{23 \times 0.7854\,n\,\varphi_n\,d_n^2\,\varphi_k}{28 \times e\,\delta \times 4.5} \quad \text{or} \quad \frac{100\,(23 \times 0.7854\,n\,\varphi_n\,d_n^2\,\varphi_k)}{28 \times e\,\delta \times 4.5}\ \%$$

$$\varrho_n = \frac{0.1433\,n\,\varphi_n\,\varphi_k\,d_n^2}{e\,\delta} \quad \text{or} \quad 14.33\,\frac{n\,\varphi_n\,\varphi_k\,d_n^2}{e\,\delta}\ \%$$

Strength of the joint.

13) The lower of the two values ϱ_b and ϱ_n gives the actual ratio of the strength in the joint to that of the full plate. When the

work is properly carried out this ratio comes out in general by the above formulæ, for

		Iron	*Steel*
single-riveted lap, plates from	6 to 24 mm thick	59 to 52 %,	56 to 47 %
double „ „ „ „	8 „ 25 „ „	72 „ 67 „,	70 „ 64 „
treble „ „ „ „	9 „ 27 „ „	78 „ 74 „,	76 „ 71 „
quadruple „ „ „ „	11 „ 28 „ „	82 „ 78 „,	80 „ 76 „
single-riveted double buttstrap, plates from	9 „ 27 „ „	68 „ 63 „,	65 „ 59 „
double-riveted double buttstrap, plates from	11 „ 28 „ „	80 „ 76 „,	78 „ 73 „
treble-riveted double buttstrap, plates from	13 „ 30 „ „	84 „ 82 „,	82 „ 79 „
quadruple-riveted double buttstrap, plates from	14 „ 32 „ „	86 „ 85 „,	85 „ 83 „

With skilful disposition of the rivets and suitable diameters a strength of 90 % and above may be obtained with a double buttstrap and a sufficient number of rows of rivets.

14) **c. The diameter of rivets** d_n depends upon the thickness δ of the plates. **Rivet diameter by Board of Trade Rules.** For thin plates d_n is not to exceed $2\,\delta$ and for the thickest plates it is not to be less than δ. It is determined by equating the shearing strength of all the rivets in one pitch to the strength of the plate between the rivets. Bearing in mind that the shearing strength of the rivet material is lower than the tensile strength of the plate, we must make the rivet section greater than that of the plate in a corresponding ratio (compare § 61, 14). Many engineers however make both sections equal, because the plate is exposed to corrosion but the rivets are not. According to 12) the rivet section to be taken into account for one pitch is

$$n \frac{\pi}{4} d_n^2 \varphi_n.$$

The length of the plate section between 2 rivets is $e - d_n$ consequently the sectional area is $(e - d_n)\,\delta$ and

$$n \frac{\pi}{4} d_n^2 \varphi_n = (e - d_n)\,\delta.$$

To compute d_n from this equation, we must eliminate e the unknown pitch. It may be replaced by the percentage of strength ϱ_b, for as the strength of strips of plate of equal thickness varies as their breadth, if we put the breadth of the full plate from centre to centre of rivets i. e. the pitch $e = 100$ we can express the breadth between the rivets $e - d_n = \varrho_b$ as $100 - d_n = \varrho_b$ and thence follows

$$d_n = 100 - \varrho_b.$$

Substituting these values in the above expression we have

$$n \frac{\pi}{4} d_n (100 - \varrho_b) \varphi_n = \varrho_b \delta$$

$$d_n = \frac{\varrho_b \delta}{0.7854 (100 - \varrho_b) n \varphi_n} \text{ cm} \quad \ldots \ldots \ldots (299)$$

The Board of Trade uses this formula for iron plates with iron rivets. For steel plates with steel rivets the expression changes as explained in 12) to

$$d_n = \frac{4.5 \times 28 \varrho_b \delta}{23 \times 0.7854 (100 - \varrho_b) n \varphi_n \varphi_k} = 7 \frac{\varrho_b \delta}{(100 - \varrho_b) n \varphi_n \varphi_k} \text{ cm} \quad . . (299^a)$$

Examples of rivet diameters.

15) Substituting in Eq. (299) for lap-joint $\varphi_n = 1$, for double-butt-strap joint $\varphi_n = 1.75$, and of course making n the number of rivets in the pitch 1 for single-riveted lap, 2 for double-riveted, 3 for treble-riveted, and 4 for double-riveted double buttstrap, we get the following diameters for iron rivets in iron plates. The accompanying values for steel rivets in steel plates are calculated by Eq. (299^a) upon the assumption that $\varphi_k = 4.5$. For ϱ_b the mean values given in 13) are substituted.

Iron Boilers	Steel Boilers	
$d_n = 1.62\ \delta$	$1.80\ \delta$	for single-riveted lap,
$d_n = 1.48\ \delta$	$1.70\ \delta$	„ double „ „ ,
$d_n = 1.10\ \delta$	$1.30\ \delta$	„ treble „ „ ,
$d_n = 1.10\ \delta$	$1.30\ \delta$	„ double „ double buttstrap with ordinary chain or zigzag riveting.

For treble and quadruple-riveted double buttstraps the rivet diameters come out variously according to the pitch, see Pl. 40, Figs. 7 to 12.

Rivet diameter according to Bach.

16) According to BACH*) experience shews that the rivet diameter should be

$$d_n = \sqrt{5\,\delta} - 0.4$$

for single, double, and treble-riveted lap,

$$d_n = \sqrt{5\,\delta} - 0.5$$

for single-riveted double buttstrap,

$$d_n = \sqrt{5\,\delta} - 0.6$$

for double-riveted double buttstrap,

$$d_n = \sqrt{5\,\delta} - 0.7$$

for treble-riveted double buttstrap.

It may be remarked that these formulæ for iron and steel rivets give rather larger diameters for iron rivets in thin plates with single and double-riveted lap, but for thicker plates rather smaller diameters than the Board of Trade Rule. For treble-riveted lap and double-riveted double buttstrap BACH'S diameters

*) C. BACH. Die Maschinenelemente. Edition VI. Stuttgart 1897. p. 149.

are throughout rather heavier than those of the Board of Trade. The same applies more or less to steel rivets, but as the Board of Trade Rule gives particularly large diameters for these with thick plates, BACH'S formulæ are preferable in this case. BACH has published tables for the riveting of the most usual thicknesses of plates and they can be strongly recommended for drawing-office use. TRAILL has also got out copious tables for the riveting of iron and steel boilers to the Board of Trade Rules*).

17) d. **The pitch** e comes out by the formula developed in 14) Pitch e.

$$n \frac{\pi}{4} d_n^2 \varphi_n = (e - d_n)\,\delta$$

for iron rivets as

$$e = \frac{0.7854\, d_n^2\, n\, \varphi_n}{\delta} + d_n \text{ cm} \quad \ldots\ldots\ldots \quad (300)$$

and by 12) for steel rivets as

$$e = \frac{23 \times 0.7854\, d_n^2\, n\, \varphi_n\, \varphi_k}{4.5 \times 28 \times \delta} + d_n = 0.143 \frac{d_n^2\, n\, \varphi_n\, \varphi_k}{\delta} + d_n \text{ cm} \quad (300^a)$$

The pitch e may be found more simply than by these Board of Trade formulæ if we assume as in 14) that $e = 100$ and $d_n = 100 - \varrho$, in this case ϱ is to be taken as the smaller of the values ϱ_b and ϱ_n.

We then get

$$\frac{e}{d_n} = \frac{100}{100 - \varrho}; \; e = \frac{100\, d_n}{100 - \varrho} \text{ cm} \quad \ldots\ldots \quad (300^b)$$

This gives the pitch as a multiple of the rivet diameter for iron rivets. For steel rivets the corresponding numerical values are to be multiplied by $\frac{28}{23}$, on the assumption that $\varphi_k = 4.5$, so that bearing in mind the value of φ_n and taking n at 1, 2, 3, 4, and 6 we get respectively for

Iron Boilers		Steel Boilers			
$e = 2.273\, d_n$	...	$1.866\, d_n$	with	single-riveted	lap,
$e = 3.333\, d_n$	...	$2.736\, d_n$	,,	double ,,	,, ,
$e = 3.570\, d_n$	...	$2.930\, d_n$	,,	treble ,,	,, ,
$e = 4.000\, d_n$	...	$3.284\, d_n$	,,	double ,,	double buttstrap,
$e = 5.000\, d_n$	...	$4.100\, d_n$	,,	treble ,,	,, ,, .

BACH determines the pitch at once thus

$e = 2.0\, d_n + 0.8$ cm for single-riveted lap,
$e = 2.6\, d_n + 1.5$ cm ,, double ,, ,, ,
$e = 3.0\, d_n + 2.2$ cm ,, treble ,, ,, ,

*) W. TRAILL. Boilers, their construction and strength. London 1888. pp. 151 & 300.

$e = 3.5\ d_n + 1.5$ cm for double-riveted double buttstrap,
$e = 5.0\ d_n + 1.5$ cm „ treble „ „ „ .

BACH'S pitch agrees pretty closely with the Board of Trade pitch for iron rivets of usual diameters from 2 to 3 cm, but it comes out greater, and therefore better in practice, for steel rivets than the Board of Trade desires, having regard to the high degree of safety at which it aims.

Greatest admissible pitch.

18) The greatest admissible pitch e for lap joints is 8 times the thickness of the plate or of the strap. For buttstraps with parallel edges the pitch must not exceed

$$e = C\delta + 4.1 \text{ cm} \quad \ldots \ldots \ldots \ldots \quad (300^c)$$

C being taken from the following table

Number of rivets in one pitch	Value of C	
	Lap	Double buttstrap
1	1.31	1.75
2	2.62	3.50
3	3.47	4.63
4	4.14	5.52
5	—	6.10

If the pitch thus calculated comes out more than 25.4 cm, the buttstraps must be cut to a zigzag shape (Pl. 40, Figs. 9 to 12) to obtain trustworthy caulking.

Spacing of the rows of rivets.

19) **The spacing of the rows of rivets** apart should not exceed $3.75\ d_n$, the lowest limit for it is fixed by the Board of Trade at $2\ d_n$ cm for chain-riveting, but they recommend

$$\frac{4\ d_n + 2.5}{2} \text{ cm} \quad \ldots \ldots \ldots \ldots \quad (301)$$

and for zigzag riveting

$$\sqrt{(1.1\ e + 0.4\ d_n)\ (0.1\ e + 0.4\ d_n)} \text{ cm} \quad \ldots \ldots \quad (301^a)$$

The diagonal pitch e_d in zigzag riveting must be so chosen that the portions of the buttstraps between the rivets are not exposed to any greater stress than the plate in the section weakened by the rivet-holes. This is the case when

$$e_d = 0.6\ e + 0.4\ d_n \text{ cm} \quad \ldots \ldots \ldots \ldots \quad (301^b)$$

But the diagonal pitch e_d must never be less than $2.4\ d_n$ cm. BACH determines the spacing of the rows of rivets as a function of the pitch e thus

0.8. e for double-riveted lap and chain-riveting,
0.6 e „ „ „ „ „ zigzag „ ,
0.5 e „ treble „ „ „ double riveted double buttstrap with zigzag riveting,

0.45 e for double-riveted double buttstrap and chain riveting with every alternate rivet in the back row omitted,

0.4 e for double-riveted double buttstrap and zigzag riveting with every alternate rivet in the back row omitted,

$^3/_8$ e for treble-riveted double buttstrap and zigzag riveting with every alternate rivet in the back row omitted.

These proportions are in accordance with good practice.

20) **The distance of the centre of the rivet from the edge of the plate** for lap-joint as well as from the edge of the buttstrap and the butt of the plate for double buttstrap-joint is generally made 1.5 d_n cm in accordance with practical tradition confirmed by experience, as necessary for satisfactory caulking. As the examples of riveting on Pl. 40 shew this distance must everywhere be preserved with zigzag-shaped buttstraps as well. **Distance from centre of rivet to edge of plate.**

21) **III. Welded joints.** In the eighties welded seams were much in vogue but they have now gone more and more out of use, on account of the continual rise in working pressure and because experience has demonstrated the uncertainty of welding which can only be faultlessly executed by quite exceptionally expert workmen. The Board of Trade formerly credited welded seams with 75 % of the strength of the full plate but now no longer admits them except in furnaces. The Germanischer Lloyd takes welded seams in ingot iron plates as having 70 % of strength. Welding is now only in use in water-tube boilers for the longitudinal seams of the top and bottom drums of THORNYCROFT'S and similar boilers as with these small diameters and thicknesses the weld can be executed with a certain amount of safety. The water-chambers of the DÜRR and D'ALLEST type of boiler are also welded together, but in these the weld is almost entirely relieved of pressure by the great number of stays, so that all it really has to do is to keep tight. **Welded joints.**

§ 63.

Stays.

1) The stays of marine boilers may be classified as **Classification.**

a) Longitudinal stays,
b) Screwed stays,
c) Diagonal stays,
d) Gusset stays,
e) Tube stays,
f) Girder stays.

Stiffeners.

2) a. **Longitudinal stays.** Boilers with large flat surfaces, especially box boilers (Pl. 19) require *stiffeners* as well as stays in order to withstand the steam pressure without deformation. The flat plates, in places where they are far apart, are therefore stiffened on the inside by angle or tee bars riveted on and spaced about 300 mm apart. The stays, which are mostly *round* and only made *flat* when want of space necessitates it, are either fitted through the plating and stiffeners and jointed up with internal and external nuts and washers or they are made with palms bolted or riveted to the stiffeners. In box boilers stays of this kind are fitted between the sides, the ends, the tube-plates, as well as between the crown and the bottom, — or the furnace and combustion chamber crowns according to circumstances.

Staying of cylindrical boilers.

3) Cylindrical boilers, on account of their stronger form, only require staying *longitudinally*. The stays are generally round and pass through the end plates with double nuts as before described (Pl. 42, Fig. 31). For high pressures the end-plates are stiffened in way of the stay-holes with washers riveted on outside (see § 61, 5 *et seq.* also Plates 22 to 27) and the stays are often screwed into the end-plates besides being nutted. This plan is recommended by the Bureau Veritas. Steel stays are not allowed to be welded. In calculating stays their smallest diameter is always to be taken.

Admissible stress on stays.

4) The following regulations are in force at present as to the stress on stays; § 60, 12 refers to their strength, extension, &c.

The Board of Trade allows for the stays of *flat* plates a maximum stress of

3.51 kg per sq. mm for iron stays welded or otherwise worked in the fire,
4.92 kg per sq. mm for iron stays not worked in the fire,
6.33 kg per sq. mm for steel stays not worked in the fire.

Where the staying of dished surfaces is necessary the rules allow 7.00 kg per sq. mm for iron stays welded or worked in the fire,
9.84 kg per sq. mm for iron stays not worked in the fire.

Lloyd's Rules prescribe as the maximum stress

4.22 kg per sq. mm for iron stays below 38 mm diameter and for iron welded stays of larger diameter,
5.27 kg per sq. mm for iron stays not worked in the fire and above 38 mm diameter,
5.62 kg per sq. mm for steel stays under 38 mm diameter,
6.33 „ „ „ „ „ „ „ above 38 „ „ .

By the Germanischer Lloyd's Rules the stress on welded iron stays is to be only $^1/_{10}$ and on unwelded iron and steel stays only $^1/_7$ of their tensile strength, or in other words

3.50 kg per sq. mm for welded iron stays,
5.00 „ „ „ „ „ unwelded iron stays,
6.00 „ „ „ „ „ steel stays.

The Veritas Rules give the following formula for the diameter d_a of the stay at the bottom of the thread

$$d_a = 3 + \sqrt{\frac{7.5\,q}{T}}\ \text{mm} \qquad (302)$$

in which q is the stress in kg per sq. mm
and T the tensile strength in kg per sq. mm.

If T is unknown it is assumed to be 35 kg per sq. mm for iron. For steel the manufacturer has to state a top and bottom limit for the tensile strength (and extension) of the proposed material, the bottom figure being then taken as the value of T. For iron q is put at 7.5 kg per sq. mm and for steel at the quotient of T obtained as above divided by 4.4. If the stays are not round, their dimensions are to be so arranged that the stress q does not exceed $\frac{T}{5.75}$ and their section is to be regarded as reduced by 1.5 mm in thickness over the whole periphery of the section for corrosion. For welded stays the tensile strength is assumed to be 20 % less. Steel stays may be welded when composed of quite soft material.

5) As far as the form of the boiler will allow the stays should be arranged in horizontal and vertical rows. For convenience of cleaning the pitch of stays should not be less than 35 cm; the maximum pitch is about 45 cm. The greatest diameter, only exceeded in case of necessity even with high pressures, is about 65 to 70 mm. The steam-space stays in double-ended boilers with combustion-chambers common to two opposite furnaces must not be brought too near the top rows of tubes as this interferes with the expansion of the end-plates and sometimes causes the top tubes to leak. The lowest row of stays in such boilers should therefore be at least 30 cm above the top row of tubes. **Arrangement of stays.**

6) b. **Screwed stays.** Flat plates which are close together and either parallel or only slightly divergent are supported by *screwed stays*. They are mostly of copper for copper fire-boxes, otherwise always of iron or steel and are screwed with a fine thread into both plates and then usually "nobbled" on the fire-side (see § 61, 4 to 8) or, — especially in plates under 13 mm in thick- **Screwed stays.**

ness, fitted with a nut at each end (Pl. 42, Fig. 33). Their dimensions are such that the stress on the section under the thread is the same as that given for longitudinal stays in 4). The Germanischer Lloyd limits the stress on screwed stays to $^{1}/_{10}$ of their tensile strength. Their diameter usually varies from 28 to 40 mm, the latter being only for very heavy pressures (15 atmos.) and consequently thick plates. The pitch l_2 of screwed stays is determined by Eq. (292) from the pressure, the thickness, and the constant characteristic of the method of fastening adopted. — In locomotive boilers the screwed stays are often drilled up the centre with a 3 mm hole from the outside nearly to the inner end (Pl. 42, Fig. 32) in order that a fracture may at once be evident by the leakage from the orifice.

Calculation of the section.

7) c. **Diagonal stays** are only used when absolutely indispensable. Their strength is calculated by regarding them in the first instance as if they were longitudinal stays fitted in the same positions. The section thus determined is to be multiplied by the ratio of the length of the diagonal stay to the perpendicular distance between the plates it supports. This product is the section of the diagonal stay.

Arrangement and strength.

8) d. **Gusset stays** are occasionally adopted in cylindrical boilers for the sake of a clear steam-space. They sometimes partially (Pl. 21, Figs. 1 & 2) and sometimes entirely (Pl. 34, Figs. 1 to 3) replace longitudinal stays. The gusset plates are connected by angle irons to the shell and the end plate. As they expose such a large surface to corrosion their section must be proportionately increased above what would be required for diagonal stays in their position.

Arrangement.

9) e. **Stay-tubes.** The front and back tube-plate are mostly stayed to each other by tubes of greater thickness than the plain ones. These stay-tubes which replace solid stays have the advantage of leaving the heating surface, — often so difficult to get in, undiminished. The stay-tubes are now screwed into the tube plates at both ends, expanded and beaded like plain tubes (Pl. 42, Fig. 1 the bottom tube). Sometimes they are not screwed into the front plate, but fitted with internal and external nuts. More rarely they are nutted at the back end also (Pl. 42, Fig. 7). Veritas recommends that there should be no nuts in the combustion-chamber. Lloyd's Rules put the admissible stress on stay-tubes at 5.27 kg per sq. mm. The back thread should be cut on the plain part of the tube thus reducing its diameter 1.6 mm a side, the strength being calculated under the thread. The tube should be swelled at the front end, so that the thread can

be cut and the tube enabled to pass clear through the hole in the front tube-plate. The spacing of stay-tubes can be taken without difficulty from the foregoing tables; the usual proportion is about 1 stay-tube to 4 plain ones.

10) f. **Girder stays** serve to stiffen the flat combustion-chamber crowns of cylindrical boilers as represented in Pl. 25, Figs. 3 & 4, Pl. 26, Figs. 3 to 6, &c. Over the combustion-chamber crown are placed several plate girders, fitted so as to stand on edge upon the tube-plate and combustion-chamber back plate, or in double-ended boilers on the two tube-plates. Each of these girders forms a support for several screwed stays fitted to the combustion-chamber crown. The following is the Board of Trade formula for iron girders and has been adopted by the classification societies; **Object and calculation.**

$$b = \frac{p\,(W - l)\,e\,L_1}{C\,h^2} \text{ cm} \quad \ldots\ldots\ldots\ldots \quad (303)$$

where

b = thickness of a girder in cm,
p = working pressure in atmos.,
W = longitudinal dimension of the combustion-chamber in cm,
l = pitch of the stays in a girder in cm,
e = distance of the girders apart in cm,
L_1 = length of a girder in cm,
h = depth „ „ „ „ „ ,
N = number of stays in a girder,
C = a coefficient fixed at

$$C = \frac{840\,N}{N+1}$$

for an odd number of stays in each girder and

$$C = \frac{840\,(N+1)}{N+2}$$

for an even number of stays in each girder.

For *steel girders* b may be reduced 10%. The girders should be about 4 cm clear of the crown plate and distance ferrules are fitted between this plate and the bottom of the girder. The size of stays and thickness of crown plate are calculated in the same way as for other flat plates.

Division XI.

Furnaces.

§ 64.

Fire-space.

Fire-space.

1) The fire-space is always inside and at the lower part of a marine boiler whatever fuel may be used. There are usually several fire-spaces, each separate one being called a *furnace*. Furnace plates must always be of the very best material.

Form and number of furnaces.

2) **I. The furnace tube.** In box boilers the furnaces are also of box-like form, in dry-bottomed boilers they have parallel sides and arched crowns (Pl. 19, Fig. 4), and in flat-bottomed boilers the furnace bottoms form inverted arches (Pl. 19, Fig. 2). Cylindrical boilers have cylindrical furnaces and in water-tube boilers the furnaces vary in form according to the position and arrangement of the tubes. The diameters most to be recommended for cylindrical furnaces range from 0.9 to 1.15 m. For artificial draught experience shews 1 m to be the best. As a general rule the number should be

for boilers	up to	2.75	m	in diameter,	1	furnace
„ „	„ „	4.0	m	„ „	2	furnaces
„ „	„ „	4.5	m	„ „	3	„
„ „	over	4.5	m	„ „	4	„

(Compare § 55, 17).

Different kinds of furnaces.

3) The principal descriptions of furnaces to which § 61, 17 applies are at present

a) plain furnaces,
b) „ „ with expansion rings,
c) FOX'S furnaces,
d) FARNLEY furnaces,
e) HOLMES'S furnaces,
f) PURVES'S furnaces,
g) MORISON'S furnaces.

4) a. **Plain furnaces,** as shewn in Figs. 3 and 4, Pl. 23 can only be applied with low pressures and small diameters, or if used with higher pressures they must be very short. When the thickness of a plain furnace calculated by Eq. (296) comes out over 15 mm, it is usually considered preferable to employ a thinner plate with stiffeners as described below. One objection to long plain furnaces is their want of elasticity. They expand when hot and exert a bending action upon the front plate and the back tube-plate, giving rise to leaks, especially in the saddle plate seam which is so much exposed to the flame. With regard to this seam it is advisable to flange up the furnace crown to meet the tube-plate so as to keep the rivets away from the direct impact of the flame as illustrated in Pl. 24, Fig. 4, Pl. 27, Fig. 6, &c. Only when special reasons, such as the possibility of renewing the furnace without cutting (see 10), interfere with this plan, should it be departed from, as in Pl. 24, Fig. 6. The union of the furnace with the front plate is now always effected by flanging the front plate in (Pl. 27, Figs. 2 & 4) or out (Pl. 24, Figs. 2, 4, & 6) and inserting the furnace into the flanged opening. Flanging the front plate inwards has the advantage of affording two caulking edges, flanging outwards, especially with furnaces of small diameter, enables the rivets to be better held up. In iron the longitudinal seam is best to be welded and in steel fitted with single-riveted double edge-strips; in either case the seams should be kept below the fire-bars. Plain furnaces.

5) b. **Plain furnaces with expansion rings.** To impart to plain furnaces greater stiffness against collapse as well as a certain amount of elasticity longitudinally, they are sometimes divided into several courses connected by one (Pl. 25, Fig. 4) or more BOWLING rings (Pl. 23, Fig. 2) or by one (Pl. 20, Fig. 4) or more ADAMSON rings (Pl. 21, Fig. 4). The BOWLING ring is rolled weldless out of the solid like a railway tyre and has answered well in practice although it involves two-ply riveting in the furnace crown. ADAMSON'S ring is a thin flat welded ring placed between the two flanged-out ends of the furnace plates so as to afford caulking edges inside the furnace. The rivets are certainly in a protected position in the water but the projecting flanges objectionably curtail the space over the furnace and interfere with cleaning. Besides these there are simple *stiffening rings* (Pl. 20, Fig. 4) consisting of a single or double angle-bar kept a certain distance off the furnace to give the water free access to it. With the present working pressures up to 15 atmos. plain furnaces are no longer fitted but are replaced by corrugated or ribbed ones. Plain furnaces with expansion rings.

Fox's furnace.

6) c. **Fox's furnace** (Pl. 41, Fig. 9) is one of the most popular in Germany because the firm of SCHULZ KRAUDT of Essen have brought its manufacture to complete perfection. It is not only fitted to Navy-type boilers (Pl. 21, Figs. 2 & 6) but also to single (Pl. 20, Fig. 4) and double-ended Scotch boilers (Pl. 26, Figs. 2 & 4). While rolled of even thickness throughout, its corrugated form renders it highly adapted to resist external pressure. KNAUDT'S*) experiments made in 1892 with 4 of FOX'S and 1 of PURVES'S furnaces of 95 cm internal diameter proved that the former with corrugations about 50 mm deep were very considerably superior to the latter in longitudinal elasticity. The four FOX'S furnaces, 10, 11.7, 12, and 13.6 mm thick were compressed 1 mm per m of their length by weights of 27000, 29000, 36000, and 49000 kg respectively, while PURVES'S, 12 mm thick, required 350000 kg to produce the same effect. The objection was at first made to FOX'S furnace that in the corrugations at the hottest part of the crown thick accumulations of almost irremovable scale would collect, causing local overheating. It must be admitted that when once the crown of a FOX furnace has become red-hot it is more easily driven in than that of a plain one, because the latter must stretch in the process whereas the former has material to spare. Years of experience have however shewn that these fears were exaggerated, especially since distilled feed-water has been resorted to. The patching of defective places is also not so difficult as it was originally assumed to be.

Farnley furnace.

7) d. **The Farnley furnace,** shewn in perspective in Fig. 2, Pl. 41 is an imitation of FOX'S. It has the same corrugations but instead of running perpendicularly to the axis of the furnace they pursue a helical or spiral course round it. This is stated to give greater longitudinal strength, thus rendering a furnace of the same thickness, diameter, and working pressure good for a greater length than FOX'S. These spiral furnaces have however not become popular, perhaps partly because an experiment instituted by Lloyd's Society in 1888 with two of them shewed that under about the same conditions they collapsed under the same pressure as FOX'S.

Holmes's furnace.

8) e. **Holmes's furnace,** Pl. 41, Figs. 3 & 5, consists of a number of plain portions about 40 cm long with intervening external corrugations 50 mm high. The advantage sought is that the corrugations, being directed away from the fire, may be better protected than the internal corrugations of FOX'S furnace and

*) Zeitschrift des Vereins deutscher Ingenieure. Berlin 1892. p. 1241.

that any deposit may be more easily removed from the plain as well as the elevated parts of the crown. A trial by Lloyd's Society in 1891 with two of HOLMES'S furnaces shewed a lower strength under equal conditions than the average of FOX'S or the FARNLEY furnaces tested about three years before.

9) f. **Purves's furnace**, Pl. 27, Figs. 4 & 6, was patented as early as 1880 and for a time proved a very serious competitor of FOX'S. It has plain portions like HOLMES'S, but they are only about 23 cm long and are separated by thickened ribs projecting on the outside about 25 mm above the plain part and hollowed out on the inside for a depth usually equal to the thickness of the plain part, so that the thickness at the summit of the rib is about 25 mm (Pl. 41, Fig. 6). The ribs are surrounded by water, do not project far out, and form no obstacle to the removal of scale from the crown. Experiments instituted partly by the Board of Trade and partly by Lloyd's Society in 1886, '87, and '89 shewed a much lower resistance to collapsing than that of any corrugated furnace submitted up to that time, even less than that of a plain furnace with an ADAMSON ring. A remarkable feature of one series of these tests was that the collapsing pressure *fell* as the thickness was *increased*, the pressures being respectively 37185 kg, 26456 kg, and 25145 kg for furnaces 8, 11, and 14 mm thick. The reason of this extraordinary yielding probably is that the ribs are of practically the same section whatever the thickness of the plain part, so that they stiffen a thin tube relatively better than a thick one. Besides this, experience has shewn that the stresses set up by changes of temperature in consequence of the unequal thickness of these furnaces and their low longitudinal elasticity, (see 6) are localized in certain places and have sometimes caused cracks at the roots of the ribs. Purves's furnace.

10) g. **Morison's furnace**, also manufactured in Germany by SCHULZ KNAUDT (Pl. 41, Fig. 4) is intended to combine the even thickness, elasticity, and strength of FOX'S furnace with the protection afforded to the ribs by the water in PURVES'S arrangement and at the same time to secure an easily cleaned surface. In the trials held in 1891 and '92 by Lloyd's Society and the Board of Trade this furnace shewed as high a strength as the best of FOX'S. In recent years MORISON'S furnace has been more and more used concurrently with FOX'S and appears to be displacing PURVES'S. It is usually formed at the back end on a plan proposed by ASHLIN which facilitates the removal of a defective furnace. The crown is flanged up to take the tube-plate, the lower half is not flanged but cut back at an Morison's furnace.

angle of about 120° to the centre line of furnace. On removing the rivets in the back end this furnace can be drawn out of the boiler.

Subdivision

11) **II. Furnace fittings for solid fuel.** It was remarked in § 19, 6 that coal is the prevailing fuel used in marine boilers and the furnace fittings described below are adapted to it, viz.

a) the grate,
b) the bridge,
c) the furnace-front.

Combustion-space and ash-pit.

12) a. **The grate** divides the furnace into two parts, the combustion-space and the ash-pit. In the *combustion-space* above the grate the combustion of the fuel lying upon it is accomplished. The proper arrangement of this space for the economical consumption of the fuel has been already described in § 21, 1 to 18. In the *ash-pit* is collected the unconsumable residue of the coals, — the ash, varieties of which are detailed in § 20, 54. The air necessary for combustion passes up from the ash-pit between the bars into the fire. The weight of and method of admitting the air are given in § 20, 24 to 33. The mouth of the ash-pit must be large enough to give passage to the required quantity of air at a speed not exceeding 1.5 to 1.7 m per sec. with natural draught, for which in general $^1/_5$ of the grate-area is sufficient.

Grate for natural draught.

13) With natural draught the grate usually consists of a number of cast iron bars of elongated trapezial section and not exceeding 1 to 1.25 m in length. The depth varies with the length up to 10 cm and the thickness at top from 30 to 40 mm. Chilled castings have lately been much used for the purpose, of the form shewn in Pl. 42, Figs. 23 to 28. Many warships, including the Imperial German Fleet have wrought iron fire-bars (Pl. 42, Fig. 29). The bars are supported at the front end by the *dead-plate* which is attached to the furnace-front and at the back end by the bridge-plate. When the grate is longer than $1^1/_4$ m, the bars are fitted in two, or even three lengths, the intermediate ends of which are carried by the *fire-bar bearers*. These are wrought or cast iron rectangular girders extending across the furnace and resting in sockets fastened to its sides (Pl. 41, Fig. 9). The total length of the grate should not exceed 2 m, as very long fires cannot be properly worked. A space of about $^1/_{25}$ of the length of a bar is left between the ends of the bars to allow of expansion. The bars have thickening strips at their ends so as to leave an interstice of about 13 mm clear between every two bars which suits the half or only slightly bituminous coals usually supplied to steamers. For

coal that clinkers much the interval can be enlarged but this involves the risk of losing a considerable amount of small between the bars. The ordinary thickness of a fire-bar is 32 mm with a head at each end 45 mm square. The sum of the areas of the interstices is called the *free grate-surface* and amounts to from 0.2 to 0.3 of the total *grate-surface.*

14) With forced draught a very much greater quantity of air has to be supplied to the fire than with natural draught and therefore a larger free grate surface is required. This is attained by making the bars thinner. They are usually of wrought iron and riveted together in sets of three (Pl. 42, Fig. 30). These bars, first fitted to the locomotive boilers of torpedo-boats, having answered well, were applied to the water-tube boilers of heavier war-ships when they were intended to be forced. Bars of this description are however not to be recommended for large cylindrical boilers because it is not practicable to keep a shallow layer of water in their ash-pits, as is done with torpedo-boats' and similar boilers for the better preservation of the bars. **Grate for forced draught.**

15) *The grate-surface.* The extent of the grate-surface and the strength of the draught determine between them the quantity of coal that can be burnt in unit time and therefore the resulting weight of steam and ultimately the power of the engine. In § 22, 8 will be found the weights of coal that can be burnt per sq. m of grate per hour in practice at various degrees of forcing; § 35, 8 shews the consumptions per *IHP* per hour as found by experience for extended runs at sea with the various systems of engines. It is thus not difficult to determine the number of horse-powers obtainable per sq. m of grate under different circumstances, with the further assistance of the table of trial-trip performances on p. 576. From experience gained hitherto the following powers may be confidently anticipated per sq. m of grate per hour **Extent of grate-surface.**

	In steady work at sea (Usual Average).	On trial when forced to the highest attainable degree.
In Locomotive boilers		
Triples and Quadruples of 13 to 14 atmos. working pressure,	110 *IHP*	350 *IHP*
Compounds of 10 to 12 atmos. working pressure,	100 „	220 „
In curved-tube water-tube boilers		
Triples and Quadruples of 14 to 20 atmos. working pressure,	120 „	240 ,

In straight-tube Water-tube boilers		
Triples and Quadruples of 12 to 15 atmos. working pressure,	100 *IHP*	140 *IHP*
In Scotch boilers		
Quadruples of 14 to 15 atmos. working pressure,	110 „	160 „
Triples of 10 to 13 atmos. working pressure,	100 „	140 „
Compounds of 6 to 9 atmos. working pressure,	90 „	100 „
In Navy-type boilers		
Triples of 10 to 12 atmos. working pressure,	95 „	130 „
Compounds of 6 to 9 atmos. working pressure,	85 „	110 „
In box-boilers		
Medium pressure compounds of 3 atmos. working pressure,	75 „	105 „
Low-pressure engines of 2 atmos. working pressure,	70 „	100 „
Low-pressure engines below 2 atmos. working pressure.	60 „	75 „

The reasons of the variations in capacity between the different types of marine boilers are explained in §§ 54 to 57. With the help of the above table and by comparison of the figures arrived at with those of certain existing ships as illustrated in Plates 21 to 38 the required grate surface for a proposed engine can be determined. The length of the grate is taken as the clear dimension between the edges of the dead plate and bridge-plate and the breadth of the grate as the breadth or diameter of the furnace.

Bridge.

16) b. **The Bridge** consists as a rule of a cast or wrought iron frame or foundation carrying a low wall of fire-bricks. The grate is inclined downwards towards the back at a slope of from 1 in 10 to 1 in 12, the bridge forms its back boundary and further serves to reduce the cross sectional area of the furnace available for the escape of the gases, so that they are constrained to linger in the combustion space in order to become intimately mixed with the oxygen of the air passing in through the grate, whereby the combustion is made as complete as possible. The area of the opening above the bridge should be about $^1/_6$ to $^1/_7$ of the grate surface, but not less than will admit of a man getting over the bridge into the combustion chamber for any purpose. Plates 19 to 29 exhibit various kinds of bridges adapted

to different types of boilers. In water-tube boilers which have no bridge it is as far as possible replaced by injecting jets of compressed air immediately above the fire (§ 57, 17) or covering the lower rows of tubes with fire-clay lumps (§ 57, 31). Sometimes doors (pigeon-holes) are fitted in the bottom web of the bridge-plate forming the back boundary of the ash-pit (Pl. 34, Figs. 1 & 2) and can be opened and closed from the boiler front. The object of these doors is explained in § 21, 15. Many Navy-type boilers have a hanging bridge built up between the fire-bridge and the tube-plate (Pl. 21, Figs. 2 & 6) to compel the gases to pass through the lower tubes. According to some experiments of LENTZ'S *) with locomotives a screen of this kind reduces the differences between the temperatures in the upper and lower portions of the combustion chamber. Without the screen the temperature was 91° C higher at the top than at the bottom and with the screen only from 54° to 65°. On opening the fire-door the temperature fell 45° at the top but steadily increased at the bottom. The screen is therefore an excellent protection against leaky tubes in boilers of this type.

17) c. **The furnace front** forms the front boundary of the fire. It is fitted with the *fire-door* above the grate and the *ash-pit damper* or *ash-pit door* beneath and was formerly always of cast-iron, but is now more and more frequently made of wrought iron or steel for the sake of lightness and durability. It is secured to the boiler front by studs and nuts so as to be easily removed. Furnace front.

18) *The common fire-door* (Plate 41, Fig. 9 & 10) was formerly also made of cast-iron but now wrought iron or steel is often used. It is hinged to the furnace-front and provided with a baffle-plate standing about 100 to 150 mm inwards from the door to protect it from burning. Small holes are drilled in the door admitting air to the fire and cooling the baffle-plate. The door is secured by a catch having a second notch to keep the door a little way open as is necessary with moderate firing. The door must not be made too large or it will permit the entry of too much cold air to the fire when opened (see § 21, 19). For large furnaces, as in locomotive (Pl. 29, Figs. 3 & 5) and water-tube boilers (Pl. 32, Figs. 8 & 10) it is better to fit two smaller fire-doors, which also enable the fire to be properly cleaned right out to the wings, better than a single large door in the centre does. Common fire-door.

*) Zeitschrift des Vereins deutscher Ingenieure. Berlin 1891. p. 1444.

Martin's fire-door.

19) *Martin's fire-door.* In recent years the fire-doors of many marine boilers have been fitted on MARTIN'S plan (Pl. 41, Figs. 7 & 8). This door is of wrought iron or steel and slightly dished outwards. It revolves on an axis at its top edge and is balanced with counter-weights so that it will remain horizontal when opened. It opens inwards instead of outwards and leaves the aperture perfectly clear. For slicing or cleaning fires the door only requires to be slightly opened, so that it not only admits but little cold air into the furnace but also protects the fireman from the radiant heat. The counter-weights are adjustable, enabling the door to be kept partly open when the fires are to be held back. — Special fire-doors for forced draught and liquid fuels are described in their proper places.

Ash-pit dampers.

20) *The ash-pit door or ash-pit damper* is fitted beneath the fire-door and serves to close the mouth of the ash-pit. It is often made so that the opening can be adjusted like that of a throttle valve to regulate the air-supply to the furnace. In merchant steamers the ash-pit dampers are mostly loose doors of sheet iron with two handles (Pl. 41, Figs. 9 & 10) and simply placed in front of the ash-pits (hung on the pricker-bar) to check the draught when required but at other times hung against the smoke-box doors. The pricker-bar is usually placed in two hooks below the dead-plate and serves as a fulcrum when using the pricker.

Varieties.

21) **Mechanical Stokers,** the object of which is to dispense with the labour of the fireman (compare p. 192), have been repeatedly tried in marine boilers although hitherto without enduring results. Those described below are the best known, viz.

α) HODGKINSON'S,
β) WHITTAKER'S, and
γ) LEACH'S.

Hodgkinson's mechanical stoker.

22) *α. Hodgkinson's Mechanical Stoker* (Pl. 43, Figs. 1 & 2). The coals are conducted to a shoot situated above the furnace and passing thence through two ducts one on each side of the fire-door, fall in front of a rectangular piston working in a horizontal box placed at the bottom of each of the ducts, the piston being slowly actuated by an excentric. The coal is thus forced into the coking box. In this box as well as on the dead-plate below, the coal is intended to be partially coked before the return stroke of the piston starts it on its journey along the grate. The velocity of feed of the coal can be regulated at will. The apparatus is driven by a belt and the grate is an oscillating one similar to that shewn in Pl. 25, Fig. 2.

23) *β. Whittaker's Mechanical Stoker* (Pl. 43, Figs. 3 & 4). From the shoot *A* the coals descend over the inclined plates *N* to the crushers *B* which regulate the feed, and thence to the rotating shovels B_1. Outside, on the shaft of *B* is a ratchet wheel *C* actuated by a pawl *D* on the bar *E*, the latter being held in position by the two round guides *F*. *G* is an oscillating lever suspended at the top and connected by a pin G_1 with the bar *E* and by a second pin *H* with the sheave *J*. This sheave is fitted on the shaft J_1 close against the wormwheel *K* which is driven by the worm K_1 on the driving shaft *L* worked by a belt at 400 to 500 revolutions per minute. The bar *E* receives a reciprocating motion from the lever *G* and communicates an intermittent rotatory motion to the two feed-rollers *B* alternately by means of the pawls *D* and the ratchet wheels *C* (about half a revolution per minute), thus conveying the coals to the shovels B_1. The rollers *B* can be stopped by throwing back the pawls. The inclined plate *N* has a regulating slide *P* at each side which is attached to a lever *R* outside the box. As the lever *R* is worked up or down the openings at the bottom of the shoot are increased or reduced thus regulating the supply of coal to the shovels. Inside the furnace is a hinged plate *V* against which a portion of the coal flying in from the shovels strikes and is carried to a greater or smaller distance along the grate according to the angle at which the plate *V* is placed by regulating it with the adjusting screw *W*. **Whittaker's mechanical stoker.**

24) *γ. Leach's Mechanical Stoker* (Pl. 43, Figs. 5 & 6). The stirrer *d* is fitted to prevent the coals getting jammed in the shoot. The coals pass through the sliding door *c* to the distributing slide *e*, driven by a sliding cross-head and serving to conduct the coals to the chamber on each side of the furnace alternately. They are thrown thence on to the grate by the revolving shovels *i* past the adjustable inclined plate *k*. The whole apparatus is actuated by a horizontal shaft (driven by a belt at 500 revolutions per minute) connected by worm-gearing to the crank which works the distributing slide and by spur-gearing to the excentric actuating the plate *k* which spreads the coals on the grate. **Leach's mechanical stoker.**

25) **III. Arrangements for burning liquid fuels.** Sufficient has been said in § 19, 15 to 22 about the nature of the liquid fuels used in marine boilers, the stowage of them is described in § 23, 23 to 44, and their advantages and disadvantages are discussed in § 24. Modern burners for liquid fuels are all on the spray or dust principle. The oil on entering the furnace is pulverized to the smallest possible particles by means of a jet of steam **Spray burners.**

or more rarely of compressed air, and while in a state of cloud or fog is burnt by the admission of the required air almost completely or at any rate much more completely than is at all possible with coal, because the air has access on all sides to the small spherical particles of oil. Sufficient air is in all cases drawn in by the jet; the supply of oil takes place from a tank placed above the furnace and kept replenished by either a special steam pump or one attached to the engines, from the ship's bunkers or other oil-spaces. The cumbustion is so good with these dust burners that there is scarcely any smoke from the funnel and the temperature often rises so high that the furnace plates have to be protected from the heat of the flame either by diverting it from the plates or lining the latter with some fire-proof material.

Pulverizers.

26) The subdivision of the oil is effected by pulverizers which are the characteristic part of the whole apparatus and may be classified as

a) Slot pulverizers,
b) Tube pulverizers, and
c) Tuyère pulverizers.

From among the great number of all kinds of pulverizers only one of the most recent of each of the three classes will be quoted here. The older ones have already been fully described and judged by the author. *)

Boehnke's pulverizer.

27) a. **Slot pulverizers.** One of the best slot pulverizers used in Russia on river-steamers is that represented in Pl. 44, Figs. 1 to 6, designed by BOEHNKE of Samara at the end of the eighties. The burner is composed of a central piece *a*, an outer ring *b* ground to fit it, and a similar internal ring *c*; these rings form the circular openings *d* for the oil and *e* for the steam. Steam and oil pass through the openings *f* from the conducting pipes to these circular slots. On issuing from the slots *g g* the oil is seized by the steam-jet and pulverized, producing a conical flame which evenly pervades all parts of the interior of the furnace. If obstructed the burner can be blown through by giving each of the regulating cocks half a turn inwards. The plug *o* then establishes communication by the hole *m* between the steam-passage *n* on the one side and the oil-passage *y* on the other side by the hole *k*. The half-turn of the plug *z* simultaneously shuts off the oil at *z* and opens a way up the middle of the plug to the oil-space of the burner. If however, as only quite exceptionally happens, blowing through

*) Zeitschrift des Vereins deutscher Ingenieure 1887. p. 989 et seq.

does not clear the slots, the outer ring can be quickly taken off by slackening three nuts and the slots cleared by wiping. To guard as much as possible against obstructions beforehand, filtering walls of brass wire-gauze lined with cloth are fitted in the tank at half its depth, through which the oil, warmed up to about 25° C., must pass so that only clean oil reaches the burners. The tank is placed above the boilers. The pulverizer illustrated is of the size for 200 *IHP* and has been manufactured by VOGEL of Samara for a number of river-steamers.

28) **b. Tube pulverizers.** BRANDT'S pulverizer (Pl. 44, Figs. 13 to 15), although invented so long ago as 1880 is still one of the most popular on Russian river-steamers. The steam and oil-pipes which can be closed by cocks are united in a brass chest. The oil issues through a circular slit adjustable by a cone attached to a stem and hand-wheel, and the steam through another slit concentric to and surrounding the first. Between the cone and the nozzle which is screwed upon the shell and therefore also adjustable, the mixture of steam and oil is formed and escapes in a fine conical sheaf to be consumed in the furnace. When at work the steam and oil-cocks are kept wide open, the flame being regulated by the cone and hand-wheel alone. The pulverizer at each furnace is so fitted that it can be turned round out of the furnace mouth as shewn by Fig. 14, examined, and after removal of the nozzle, thoroughly cleaned. The air necessary for combustion enters the furnace through holes drilled in the furnace front and ash-pit door. If spiral grooves are cut in the regulating cone the flame forms a vortex filling the entire furnace. **Brandt's pulverizer.**

29) **c. Tuyère pulverizers.** For the last nine years SCHICHAU of Elbing has repeatedly fitted the tuyère pulverizer illustrated in Pl. 44, Figs. 7 to 10 with the best results. This burner can be applied to boilers which burn coals also as it only requires to be bolted to the fire-door. The steam and oil are admitted through stop-valves to a tuyère-like apparatus. The section of the steam jet which is in the centre can be easily adjusted by the conically tipped spindle with hand-wheel and the amount of air carried in by the jet can be regulated by a larger hand-wheel controlling the position of the tuyère in the outer shell. By screwing back the spindle regulating the steam supply the apparatus can be blown through or when out of action the middle portion of the tuyère can be screwed out for cleaning. **Schichau's pulverizer.**

30) **Relative advantages of the pulverizers.** The slot pulverizers consume on an average 3 kg of oil per *IHP* per hour, the tube pulverizers 2 to 2.5, and good tuyère pulverizers only about 1 kg, which **Economy and trustworthiness.**

is reported to have been again reduced to about half with modern triple-expansion engines. The slot pulverizers require about 6 to 8 % of the total steam consumption of the engine to actuate them, the tuyère pulverizers about 4 to 6 %, and the latest and best of this system only about 2 to 3 %. For a new design however it is best to estimate 5 %. The greater economy of the tuyère pulverizers lies in their superior action which subdivides the oil more finely. In this respect the slot pulverizers are the least efficient. The tuyère pulverizers are also in general those least subject to obstruction, the tube pulverizers being the second best on this point and the slot pulverizers the worst. In any case an obstruction of the tuyère can generally be very easily got rid of by screwing back the steam-regulating cone, so that this system is not only the most economical but the most trustworthy in actual work. Other good examples of tuyère pulverizers are illustrated, viz. D'ALLEST'S Pl. 44, Figs. 11 & 12 and SPAKOWSKI'S Figs. 10 & 17.

Thwaite's furnace.

31) **IV. Furnace arrangements for burning gas.** In § 19, 24 to 28 it was shewn how little prospect there is of successfully using gas for firing marine boilers, several examples being quoted, especially those where the gas was distilled from liquid fuels. The reference to these experiments made in the sixties will be here supplemented by a short description of THWAITE'S proposed plan of firing with coal-gas*) (1885). He fits a range of gas generators two storeys high, athwartships, between the engine and boiler rooms. The generators are surrounded on three sides by water-jackets and at the front by air casings to reduce the radiation. The gas-pipes leading to the furnaces are lined out with fire-resisting material and pass through tanks containing feed-water which they heat and thus partially purify. The ignition flame of the gas does not come into contact with the boiler furnaces but they only receive the radiant heat of the combustion of the gas which takes place in a sort of regenerative apparatus. No soot is supposed to be deposited in the furnaces or tubes. The regenerative apparatus consists of a grate of firebricks which are brought by the passing gases to a white heat. Unfortunately masonry of any kind about furnaces at sea has hitherto never been a success, and this is probably the cause of THWAITE'S failure, although the firing of marine boilers with gas must be regarded as an ideal to be striven for. (Compare § 21, 27.)

*) Iron. 1885. Vol. 26, p. 347.

§ 65.
Flues.

1) **I. The Flues** or passages traversed by the gases in box boilers and Scotch boilers may be subdivided into — Subdivision.

a) the combustion-chamber,
b) the tubes,
c) the smoke box,
d) the uptake, and
e) the funnel.

Locomotive and Navy-type boilers have no combustion-chamber and the smoke box is situated at the back end of the boiler (Pl. 20, 21, 28, & 29). Water-tube boilers often have separated flues among the tubes, formed in straight-tubed boilers by plates or fire-brick divisions (Pl. 30, Figs. 1 & 14 and Pl. 31, Fig. 8) and in curved-tubed boilers by placing certain portions of the tubes close together (Pl. 32, Fig. 8 and Pl. 33, Fig. 4). From these passages the gases escape direct to the uptake and funnel.

2) **a. The combustion-chamber** conducts the gases coming over the bridge into the tubes. It has been stated in § 55, 18 and 21) that a separate combustion-chamber for each furnace is preferable to one common to two or more furnaces. Care should be taken not to design the combustion-chamber too narrow (longitudinally to the boiler), as the comparatively great weight of roomy combustion-chambers is more than compensated for by the improved combustion and increased facility for repairs they afford. Separate combustion-chambers should not under ordinary circumstance measure less than 70 cm longitudinally to the boiler and in double-ended boilers where one common combustion-chamber is fitted to two opposite furnaces it should measure at least 120 cm between the tube-plates, so that cold air cannot pass through an open fire-door and striking directly against the opposite tube-plate thus render the tube ends leaky. The shape of the combustion-chamber depends upon the design of the boiler as Plates 22 to 27 shew. To assist the circulation it is well to make the combustion-chamber sides diverge upwards with regard to the boiler shell and to keep the water-spaces at least 15 cm wide at the narrowest parts. The thickness of combustion-chamber plates is found from Eq. (292) but the bottom is nearly always made thicker than the sides to provide against the greater corrosion to which it is exposed. The method of uniting the plates is to flange the tube-plate and back and fit the crown, wrapper, and bottom over them. — Construction.

Staying of combustion-chamber crowns.

3) An interesting system of staying combustion-chamber crowns is adopted in the boilers illustrated in Pl. 27. It is taken from locomotive boilers (Pl. 28 & 29), only that here jointed stays instead of fixed stays are fitted on account of the cylindrical boiler crown. When this system was first introduced in 1890, MARTIN (the inventor of the fire-door before described) asserted *) that it was only applicable to open-bottomed locomotive boilers although the portable stays possessed an advantage over the usual girder stays for convenience of cleaning, further that in boilers with closed combustion-chamber bottoms like those illustrated, the outside rows of the crown stays might be exposed to compression and therefore be too weak although possessing a factor of safety of 7. To get over this compressive action he recommended staying the combustion-chamber bottom as shewn in Pl. 27, Fig. 6. He does not consider the stays in the combustion-chamber sides, assisted by the resistance of the furnaces and tubes to bending upwards and further reinforced by the weight (minus the displacement) of these parts, sufficient. However the experience gained with numerous successful boilers (Pl. 27, Figs. 2 & 4) shews that there is no need to stay combustion-chamber bottoms.

Tubes.

4) b. **The tubes** always provide the greatest part of the heating surface of a boiler; they are distinguished by

- α) material,
- β) dimensions,
- γ) arrangement in the boiler,
- δ) method of jointing,
- ε) stoppers.

Material.

5) α. **Their material** is now either steel, iron, or brass. Steel tubes have latterly been made of soft material of about 40 kg per sq. mm tensile strength and fitted to boilers almost all of over 10 atmos. working pressure. Iron tubes are made of the best soft malleable iron, lap-welded, and are more rarely used since the introduction of drawn steel tubes. Brass tubes are about four times as expensive as iron ones but they last twice as long and when they are condemned are worth about half their original cost, so that considering their high conductivity as well, they are in fact more economical than iron or steel tubes. The disadvantage of them is that their original comparatively low strength is very much more reduced by the high temperatures of high-pressure boilers than the strength of the other tubes is, so that brass tubes are only fit for low pressures.

*) Engineering 1890. p. 228 et seq.

6) *β.* **The dimensions of the tubes** depend upon the working conditions of the boiler. War-ships as a rule have tubes of smaller diameter than merchant ships as the former are required to have a particularly large heating surface; the usual diameter in the Navy is 63.5 mm for Scotch boilers, sometimes rising to 70 or falling to 57 mm; in locomotive boilers it varies between 44 and 52 mm. Scotch boilers in merchant steamers however have tubes 75 to 100 mm diameter, the most usual sizes being 76, 83, and 89 mm. But with forced draught, where the conditions therefore more resemble those of the Navy, diameters as low as 63.5 mm are adopted. The thickness of the tubes varies with the diameter and pressure whatever material they may be made of. Their length with the ordinary grate 2 m long is generally 24 to 25 external diameters in Scotch boilers; in short boilers with large tubes it comes down to 20 diameters and rises to 30 in long boilers with narrow tubes. In locomotive and Navy-type boilers the length is increased to 35 diameters and in boilers always worked under artificial draught lengths of 40 to 60 diameters occur. **Diameter and length.**

7) A peculiar kind of tube with internal ribs (Pl. 42, Figs. 12 & 13) has been introduced by SERVE and frequently adopted in recent ships as in the boiler Pl. 24, Figs. 5 & 6. It is stated that the conductivity of the ribbed tubes is greater than that of plain ones and some evaporative trials held in France*) are reported to have shewn a saving of 20 % of coal which of course is exaggerated. Further experiments were made in England**) and left no doubt as to the economical superiority of the SERVE tube. The ribs are stopped at the ends so that the tubes can be expanded in the usual way. These tubes have also been fitted to water-tube boilers as they resist sagging better than plain tubes (compare § 57, 31). An objection to SERVE tubes is the difficulty of cleaning them, a brush of special form being required. In order to more completely absorb the heat of the gases spiral retarders are often inserted in the tubes, consisting of twisted strips of hoop iron having about 2 to 3 turns in the length of the tube (Pl. 42, Figs. 10 & 11) and thus constraining the gases to traverse a longer path on the inside surface of the tube. This has such a powerful cooling effect on the gases that according to some experiments by GEBHARD of Brest***) their temperature on leaving the tubes is only 0.6 of what it is without retarders. Those used in the trials were **Serve tubes and retarders.**

*) Engineering. 1889. I. p. 288.

**) Engineering. 1890. II. p. 490 and The Marine Engineer 1889. p. 116.

***) A. BIENAYMÉ. Les machines marines. Paris 1887, p. 479.

6 cm wide and of 36 cm pitch. In boilers with comparatively weak draught the retarders have had to be abandoned as they diminished the activity of the combustion. Another device for rendering the contact of the gases and the tube surfaces more intimate is the so-called "Cerberus" ferrule illustrated in Figs. 8 & 9, Pl. 42. LEWIS of Wolverhampton has lately introduced tubes similar to the FARNLEY furnace (Pl. 41, Fig. 2) with spiral grooves which the flame is supposed to follow. These tubes are stronger and more elastic than ordinary plain tubes.

Spacing of tubes.

8) γ. **The arrangement of the tubes** in Scotch and Navy-type boilers is always in horizontal and vertical rows and the usual pitch is 1.4 times the diameter of the tubes in both directions. But the British Admiralty requires 29 mm space between tubes of 63.5 mm diar. in Scotch boilers, that is about 1.45 times the diar., while some designers go as far as 1.5 times the diar. In locomotive boilers however the tubes are placed as shewn in Pl. 28 & 29 in zigzag rows because it is always a necessity to get in the largest possible heating surface. The zigzag rows generally enclose an angle of 60° and the diagonal pitch of the tubes is sometimes as low as 1.3 times the diar. Both for cleaning and to diminish priming it is advisable to keep the tubes in parallel vertical and horizontal rows and as far apart as can be managed. The zigzag arrangement hinders the ascent of the steam-bubbles. The consequence is that the bottoms of all the tubes are covered from end to end with steam-bubbles and therefore are hotter and expand more than the tops from which the bubbles are more easily liberated. This continued uneven expansion causes the tubes to sag and retain a set after they are cold which amounts sometimes to 13 mm in brass tubes 2.5 m long and 5 or 6 mm in steel tubes of the same length. The sagging becomes worse when the boilers are forced and slackens the ends in the tube-plates. The slackening is aggravated in locomotive boilers by the extreme heating of the tube-ends and tube-plates occasioned by the accumulation of bubbles prevented by the zigzag arrangement from getting away fast enough. To facilitate the access of water to the necks of the tubes SCHICHAU spaces them in locomotive boilers for torpedo-boats so that the vertical rows diverge upwards somewhat.

Methods of jointing the tube-ends.

9) δ. **The jointing of the tube-ends** in the tube-plates is effected by an expander which virtually enlarges the outside periphery of the tube in the hole. To improve the joint the projecting ends (at the back) are sometimes beaded. It is advisable to have the front ends of the tubes swelled 1 or 2 mm larger in diameter than the

plain part (Pl. 42, Fig. 1), so that they may be renewed without much trouble; tubes thus adapted are supplied at very little extra cost over perfectly plain ones. The tubes of locomotive boilers are screwed into the front tube plate and afterwards expanded and beaded. Where the thread is to be cut, the tube-end, whether of iron or steel, is jumped and sometimes iron tubes are strengthened by having a liner welded inside (Pl. 42, Fig. 4). In the back tube-plate they are expanded in the usual way, but brass tubes are also screwed into the back tube-plate, expanded and beaded. If the tubes of a new high-pressure boiler leak, insufficient expanding in the first place is often the cause, or else that they have lost their tension through over-heating. In either case the remedy is to re-expand them as much as possible which cures or greatly mitigates the leakage. Sometimes a want of care in boring the holes in the tube-plate occasions leaky tubes in new boilers. Variations of 1 to 1.5 mm have been found in the diameters of the holes in tube-plates and some holes nearly as much as 1 mm oval. A special method of jointing by means of a cone welded on each end of the tube has been adopted by PAUCKSCH (Pl. 42, Fig. 3) and has answered well for smaller boilers of not too high pressure, especially auxiliary boilers (Pl. 34, Fig. 5). The tube-plate holes are bored taper and the tubes driven tight into them. They can be easily driven out again when the boiler is to be thoroughly cleaned as the small end of the front hole is rather larger than the big diameter of the back-end cone. BERTIN*) describes a third kind of tube joint, that of CARAMAN, adopted in the French Navy. Two grooves are cut half in the periphery of the hole and half in the tube and a thin electrum or brass wire ring is slipped into each of them, upon which the tube is subsequently expanded. This method is tedious and costly and is also applied to water-tube boilers but the result is no better than with the common system of expanding.

10) **Ferrules** of wrought iron, steel, cast-iron, or best of all, malleable cast-iron are sometimes driven into tubes which are past expanding and have their beading burnt off. These rings which are somewhat tapered on the outside (Pl. 42, Fig. 5) certainly improve the tightness of the tubes, but they contract their sectional area and favour the accumulation of soot inside them. For locomotive boilers, the tubes of which very often become leaky during heavy forcing, the British Admiralty has introduced with much success the cap ferrule**) shewn in Pl. 41, Fig. 6 **Ferrules.**

*) L. E. BERTIN. Chandières marines. Paris 1896. p. 187.

**) Transactions of the Institution of Naval Architects. London 1893. p. 146.

the satisfactory action of which is chiefly ascribed to the badly conducting layer of air enclosed between the ferrule, the tube, and the tube-plate. Malleable cast-iron has proved to be the best material for these ferrules.

Tully's tube expander.

11) **Tube-expanders** have been produced in many varieties; the underlying principle of them all is that 3 or 4 rollers are pressed against the inside of the tube by a tapered drift. One of the latest and best is TULLY'S (Pl. 42, Figs. 19 to 22). The hollow cylinder *a* is constructed in one piece with the ratchet *b*, the pawl lever *c* being fitted on it and kept in place by the loose collar *d*. The sleeve *e* is inserted into the cylinder *a* and has a groove into which the screw *f* enters to prevent the sleeve from revolving in the cylinder. The screwed spindle *g*, which can be rotated but not shifted end-wise in the cylinder *a*, gears into a female screw at the back end of the sleeve *e* and presses the sleeve against the end of the drift *h* so as to drive it forwards. When the spindle *g* is screwed back the screw *s*, engaging in a groove in the drift, carries the drift with it. The distance ring *k* can be set at the required position on the roller shell by means of pinching screws. In this expander the point of attack of the force actuating the rollers, that is the ratchet lever, is closer up to the rollers than in any other apparatus of the sort which gives it a great superiority in handiness, and the rotating movement of the drift is made entirely independent of that of the rest of the tool whereby the friction is considerably diminished. This expander is therefore very rapid and easy in action.

Varieties.

12) ε. **Tube-stoppers** are indispensable on board steamers as defects in tubes are of not exactly rare occurrence. The Germanischer Lloyd and with it the German Seeberufsgenossenschaft or Nautical Association in their Rules for spare-gear prescribe 1 tube-stopper for every 50 tubes fitted. All tube-stoppers are so arranged that they can be inserted into the defective tube from the front and afterwards close both ends of it. Of the many varieties of tube-stoppers only the best-known will be mentioned here, viz.

HOUILLE'S,
SIMONY'S, and
LATIL'S.

Houille's tube-stopper.

13) *Houille's tube-stopper* *) (Pl. 42, Fig. 17) is to some extent formed on the plan of a stuffing-box. At each end of the tube are two cast-iron discs which compress a thick asbestos washer

*) L. E. BERTIN. Chaudières marines. Paris 1896. p. 191.

between them, the two inner discs abutting against a distance tube and the two outside discs being drawn towards the inner ones by the nut at front end. The distance tube is kept central to the bolt by a plate washer driven into it at each end.

14) *Simony's tube-stopper* (Pl. 42, Fig. 18) is similar in principle to the above except that the asbestos washer is replaced by a metallic joint formed by two conical rings fitting over each other at each end of the tube. The inner ring is harder than the outer one and when the nut at the front is tightened up the outer ring is pressed against the inside of the tube, forming a perfectly tight joint. **Simony's Tube-stopper.**

15) *Latil's tube-stopper* (Pl. 42, Figs. 14 to 17) has a folding disc at the back end which, as figures 16 & 17 shew, is pushed through the tube from front to back. As soon as this disc is clear of the back end of the tube it opens out by its own weight and can be drawn against the tube end by the front nut. The joint is formed by asbestos rings let into the discs. To form a provisional joint while tightening up the nut an india-rubber washer is fitted between two smaller iron washers just behind the front disc. **Latil's Tube-stopper.**

16) **The tubes** of water-tube boilers are also made of steel, iron, brass, or copper. With the latter materials however the experience has not been fortunate (compare § 57, 64), so that the comparatively small tubes of curved-tubed boilers are mostly made of steel, while for the larger tubes of the water-chamber boilers iron is still frequently used. In war-ships the tubes used to be galvanized on both sides to increase their durability, but since it has been found that internal galvanizing causes trouble (compare § 57, 59) it has been abandoned and the tubes are galvanized externally only. Curved tubes are usually secured in the tube-plates by expanding in the ordinary way, the projecting ends being afterwards opened out by drifting. To give a better hold to the tubes in the comparatively thin tube plates, grooves are cut in the tube holes like a very coarse thread into which the material of the tubes (only about 2 mm thick as a rule) is forced by the expander. Some designers, for instance DU TEMPLE, do not drift open the projecting ends but screw them and fit nuts to prevent the tubes drawing. REED secures his tubes with nuts only as shewn in Pl. 39, Fig. 18. The straight tubes of DÜRR'S and NICLAUSSE'S water-chamber boilers are held entirely by a conical joint illustrated on Pl. 39. In boilers with two water-chambers the tubes are simply expanded. Pl. 39 shews a number of joints for the tube-end doors in these water-chambers. **Tubes of water-tube boilers.**

Arrangement. 17) c. **The smoke-box** is in box boilers built into the boiler, but in Scotch, locomotive, and Navy-type boilers it is constructed of iron or steel plates about 2 to 5 mm thick connected by angles and studded to the boiler front. Single-ended boilers which work under natural draught only have one smoke-box common to all the furnaces but in boilers with artificial draught the smoke-boxes are often separated from each other. The smoke-box of a cylindrical boiler is about 30 to 40 cm deep at the bottom, increasing to about twice that depth at the top. The sides and bottom are kept about 50 to 60 mm clear of the outside rows of tubes for convenience of handling the tube-brushes. For sweeping tubes the smoke-box is provided with doors at the front generally made to open sideways, sometimes, but more rarely, upwards. The doors are secured by catches and are usually made of three plates, the middle one being the actual door and thickest, the other two being internal and external baffle plates. Figs. 7 to 10, Pl. 44 illustrate the whole arrangement for a Scotch boiler.

Uptakes. 18) d. **The uptake** is placed upon the smoke-box and serves to conduct the gases from a group of boilers to the funnel. The uptake varies very much in form with the arrangement of the boilers connected to it and should have no sharp corners but easy flowing lines so as not to interfere with the free current of the gases. This is best attained and the draught in the fires sharpened if the uptakes from the several boilers are separated by division plates continued up to the base of the funnel. The plates and angles of the uptake are usually of the same scantlings as those of the smoke-boxes. The top front plate of a Scotch boiler should be protected in the uptake by a baffle-plate.

Arrangement. 19) f. **The funnel** carries off the gases from the uptakes to the open atmosphere. Each group of boilers has its own funnel, so that some ships have 2, 3, and even 4 funnels. The above-mentioned division plates in the uptakes must only be omitted when the draught from all the boilers to one funnel runs in the same direction; if any of the draught currents in the uptakes are opposed to each other the division plates must be carried well up into the funnel to prevent the currents baffling, but it is better for the divisions to extend to the top of the funnel. Well-designed funnels often have, besides the above, an air shaft up the centre or on one side which extends down to the boiler tops for ventilation. To regulate the draught and to shut off boilers not at work *funnel-dampers* are sometimes fitted (Pl. 45, Fig. 3). Superheaters are often placed round

the bases of the funnels of box boilers if they are not fitted in the uptakes.

20) The draught-producing action of the funnel is explained in § 21, 47, — it is the more effective **Action of the funnel.**

a) the greater the difference of pressure above and below the grate, i. e. the higher the funnel; but in very high funnels the frictional resistance to the ascent of the gases is greater than in low ones and this circumstance limits the effect somewhat;

b) the smaller the weight of the gases in the funnel, i. e. the less their tension, or in other words, the higher their temperature; but a very high funnel temperature entails considerable losses of heat;

c) the lower the conductivity of the funnel walls, protecting the gases against cooling in the funnel; with this object double funnels are now often fitted;

d) the smoother and cleaner the internal surface of the funnel; a considerable deposit of soot increases the frictional resistance.

21) The height of the funnel above the bars is considerably limited in marine boilers and depends to some extent upon the dimensions of the ship. It varies from 2 m in launches to a little over 20 m in very large seagoing steamers and only reaches 30 m in the largest mail-steamers. In small craft like gun-boats, torpedo-boats &c. some arrangement to assist the draught has often to be fitted in order to get the necessary air for the fires with such short funnels, as further described in §§ 66 & 67. It may here be remarked that all these devices effect either a rarifaction of the gases in the funnel or an overpressure or plenum beneath the bars, both of which vary between 0.001 and 0.015 atmospheres (about 10 to 150 mm of water). To dispense with any apparatus of this sort the funnels of large swift steamers have of late years been made as high as 36 m, for instance in the "Scot" by DENNY, where a draught of 16 mm of water is stated to be obtained. Since this ship came out fast cruisers have received considerably higher funnels than formerly and many have had their funnels heightened, for every meter in the height of a funnel of ordinary design is considered equal to 1 mm of water pressure in the draught at the fires. **Height of funnel.**

22) Funnels may be distinguished as **Varieties of funnels.**

α) fixed funnels,

β) hingeing funnels,

γ) telescope funnels.

Fixed funnels. 23) α. *Fixed funnels* are the most usual, all merchant steamers and sailless war-ships have them (Pl. 45). The section is generally circular, but very large ones are sometimes made elliptical or with flat sides and rounded ends to diminish the air resistance and save breadth in the boiler casing. From considerations of strength, the plating at the foot of the funnel must be thicker than at the upper parts although these are more exposed to corrosion. The plating is fitted with edge-strips to give a neat and smooth exterior. Large funnels are stiffened inside with angle or channel bars. Some distance from the top (Pl. 45, Figs. 3 & 4) eyes are riveted on to take the stays (chain or wire rope) to support the funnel against wind pressure. The stays must not be tautened up until the funnel is hot. Many funnels have jewel-blocks fitted for hoisting a stage for the purpose of painting. Steps are sometimes fitted inside the funnel for cleaning. When the boilers are laid off for a considerable time the funnel should be closed at the top by a sheet-iron cap to keep out the rain. The arrangement, on board the U. S. protected cruiser "Columbia", of the auxiliary boilers within the main boiler casing (Pl. 45, Figs. 1 & 2) is remarkable.

Hingeing funnels. 24) β. *Hingeing funnels* are only fitted to river steamers requiring to pass under bridges and were adopted in the small obsolete rigged cruisers whose depth of hold was insufficient to accommodate telescope funnels. Hingeing funnels of considerable dimensions require a winch if they are to be smartly handled.

Telescope funnels. 25) γ. *Telescope funnels* are scarcely ever fitted now and are only found on the old full-rigged cruisers. Handy as the telescope funnel is when under canvas, allowing the main course to be carried and diminishing the air resistance when by the wind, these advantages are counterbalanced by its drawbacks when under steam. The space between the standing and travelling portions always admits cold air which spoils the draught. How seriously this reduces the boiler power may be inferred from the fact that the Bureau of Steam Engineering of the U. S. Navy recommended the abandonment of telescope funnels in 1873 in spite of their advantages under sail, so as to help up the speed of the war-ships when under steam.

Funnel winch. 26) The travelling part of a telescope funnel is hoisted or lowered by two chains, one on each side, passing over sheaves fitted in the boiler casing and taken to a hand winch placed in the tweendecks worked by worm-gearing. Four guide rollers are fitted at the heel of the telescoping part. When the funnel is hove up it rests upon 4, 6, or 8 iron wedges (according to its

size) which slide in guides attached to the funnel-casing. Stays are fitted as for fixed funnels and a rack and pall is provided on each side to guard against the chains breaking during hoisting or lowering.

27) Besides the air-casing a second iron screen or trunk surrounds the funnel at a greater distance off. This trunk, called the boiler hatch or fiddley, fulfils the double purpose of protecting the adjacent portions of the ship from radiation and forming a down-cast for the air to the fires. The boiler hatch terminates in a coaming on the upper deck which usually has sliding shutters in the sides and gratings on the top arranged with covers to be battened down in bad weather. Boiler hatch.

28) **II. The heating surface** is the area of the plates, tubes, &c. in contact with water on one side and fire on the other. In tubular boilers the heating surface therefore includes Heating surface.

1. The furnace crowns, i. e. all above the bars,
2. the combustion chamber „ „ „ „ ,
3. the tubes,
4. the front tube plate up to the water line.

The surfaces exposed to the action of the flame, 1 and 2 are frequently called the *direct heating surface*, 3 by itself the *heating surface in the tubes*, and 3 and 4 together the *indirect heating surface*, because it is removed from the action of the flame. The tube-surface is always the largest portion, about 80 % of the total. In England the front tube-plate is now not generally taken into account. At the British Admiralty the back tube-plate is also omitted but to make up for it, the length of the tubes is measured from outside to outside of tube-plates, to simplify the calculation, the result being practically correct. In water-tube boilers it is usual to regard as heating surface only that of the submerged portion of the tubes, but we may also include

1. As much of the surface of the bottom drums as is accessible to the action of the flame in boilers of THORNYCROFT type (Pl. 32, Figs. 3 & 8),
2. the tube-plates up to the water-line in boilers of D'ALLEST or DÜRR type (Pl. 31, Fig. 9 and Pl. 30, Fig. 13),
3. the surface, up to the water-line, of the top drums with drowned tubes, in boilers of NORMAND type (Pl. 32, Fig. 10).

The top drums do not otherwise count as heating surface as they are usually employed as feed-heaters.

29) **The action of the heating surface** depends upon Action of the heating surface.

a) its extent, quality, and condition as to cleanliness on the fire and water sides,

b) its form, position, and arrangement,
c) the difference between the temperatures on the fire and water sides,
d) the time during which the transmission of heat can proceed,
e) the conductivity of the material forming the heating surface and how the heat is communicated, whether by flame, hot fuel, or combustion gases.

Condition of the heating surface. 30) a. **The condition of the heating surface** as regards cleanliness is of importance inasmuch as boiler scale and soot are both very bad conductors of heat and may entirely cripple the boiler. The evaporation of a clean boiler is therefore always higher than that of a dirty one.

Position of the heating surface. 31) b. **The position of the heating surface** has a great influence upon its efficiency as the following experiment of ARMSTRONG'S shews. If a hollow metallic cube be filled with some highly heated substance and suspended in water so that two of its faces are about level, more than twice as much water will be evaporated per unit area of the top face than of the vertical faces and there will be no evaporation on the bottom. This is explained by the difficulty with which the steam-bubbles liberate themselves from the vertical surfaces to give place to fresh water-particles, so that a thin layer of badly-conducting steam is formed between the hot vessel and the water. The steam generated on the bottom surface cannot get away at all. This view is corroborated by suspending the hot cube obliquely (by one corner) in the water, upon which the two faces that before were vertical and are now directed obliquely upwards shew a greater evaporation than before, whereas the two faces now directed downwards shew considerably less. Hence it is to be inferred that in a tubular boiler the furnace crowns are the most effective portion of the heating surface, then come successively the haunches of the furnaces above the bars, then the combustion-chamber crowns, backs and sides, the tubes and the front tube-plates. How remarkably the effect of the hot gases falls during their course along the tubes has been demonstrated by repeated experiments. In a tubular boiler divided internally into six water-tight compartments the following quantities of water were evaporated during a three hours' trial, viz.

in the	back	compartment	25	mm	long	1.71	kg
„	„ 2^{nd}	„	254	„	„	1.75	„
„	„ 3^{rd}	„	305	„	„	1.11	„
„	„ 4^{th}	„	305	„	„	0.82	„
„	„ 5^{th}	„	305	„	„	0.67	„
„	„ 6^{th}	„	305	„	„	0.63	„

The high evaporation in the first compartment, only 25 mm long, is doubtless due to the action of the tube-plate and a comparison of the second compartment with the others shews the rapid fall in the efficiency of the heating surface from back to front. Another experiment was made by GRAHAM with three cylindrical vessels, one, A, placed over a furnace, the second, B, at the side of it, and the third, C, also at the side of, but further off the fire. The following rates of evaporation per sq. m of heating surface were observed.

A	74.5 kg,	100 %,
B	25.85 „	35 „ ,
C	11.92 „	16 „ ,

affording further evidence of the rapid fall in efficiency of the heating surface as well as of the great importance to be attached to the "direct" portion of it. COUCHE states (§ 56, 11) that in locomotive boilers 1 sq. m of fire-box surface is worth about 6 of tube-surface and according to the experiments of LENCAUCHEZ and others the evaporation per sq. m per hour on surfaces exposed to the beat of the flame is about 2.5 times as great as the average of the entire heating surface.

32) c. **The difference of temperature** between the feed-water and the combustion-gases must be as small as possible, and the hotter the water is when entering the boiler the greater the efficiency of the heating surface becomes. In order therefore to get this difference of temperature as low as possible the feed is introduced at the coldest part of the boiler where the combustion-gases are most cooled down and the water is allowed gradually to travel to the hottest parts of the boiler. Difference of temperature.

33) d. **The time** during which the communication of heat is proceeding must be as long as possible for the heat of the gases is thus more completely taken up, so that in general a boiler with large heating surface is more economical than one with less. In the former case however the steam produced per unit area of heating surface in unit time is less than in the latter, that is the boiler steams more slowly. Time.

34) e. **The conductivity** of copper and brass surfaces is greater than that of iron and steel ones, the former metals are therefore sometimes employedfor fire-boxes and tubes to render the heating surface as efficient as possible and when this is done the fire-box and tube surface together can in general be made 30 % smaller for a required evaporation than when iron or steel is employed. Conductivity.

35) **The extent of the heating surface** varies with the type of boiler and engine. In general it ranges between 0.3 and 0.4 sq. m per *IHP* with natural and from 0.15 to 0.2 sq. m with artificial draught. Averages.

Column 45 of the tables on pp. 594 to 601 shew that the average proportions with natural draught are

0.33 sq. m per *IHP* in box, navy-type, and Scotch boilers for low-pressure and compound engines,

0.25 sq. m per *IHP* in Scotch boilers with triples,

0.20 „ „ „ „ locomotive boilers with triples.

For quadruples we may go as low as 0.22 sq. m per *IHP* in Scotch boilers. The British Boiler Commission already referred to recommends 0.233 sq. m for Scotch boilers with triples. Experience shews the heating surface of water-tube boilers to be less efficient than that of tubular boilers, so that to be on the safe side with these boilers we must allow 0.30 sq. m of heating surface per *IHP* for triples and quadruples. Water-tube boilers under forced draught (combustion per sq. m of grate from 120 to 135 kg of coal per hour) have produced on an average 1 *IHP* with from 0.23 to 0.24 sq. m of heating surface with triples.

Velocity of evaporation. 36) *The velocity of evaporation* of a marine boiler is the weight of water evaporated per sq. m of heating surface per hour. It varies with the character of the heating surface and the draught between 25 and 50 kg. The lower figure applies to the older box boilers when not perfectly clean, the latter to torpedo-boat boilers with copper fire-boxes and artificial draught. Good water-tube boilers have evaporated as much as 40 kg of water per sq. m of heating surface per hour.

Evaporative capacity. 37) *Evaporative capacity* means the total weight of water evaporated by the boiler per hour. The idea of *evaporative power* is explained on p. 167.

Numerical values. 38) **The ratio of heating surface to grate** is given for the various types of mercantile and naval boilers in the tables on p. 403, col. 15, p. 570, line 7, pp. 597 and 601, col. 47. With natural draught the heating surface is about 25 to 30 times the grate surface, with forced draught it rises to from 40 to 60 times the grate. The British Boiler Commission (see note at foot of p. 525) recommends a ratio of 33 : 1 for Scotch boilers with natural draught.

Clark's investigations. 39) CLARK carried out a series of searching experiments on the ratio of heating surface to grate and the consumption of coal and water with locomotive boilers, the results of which are in general applicable to marine boilers also and may be collated as follows.

a) For a given heating surface the economy of fuel falls in the same proportion as the grate-surface is increased. If therefore the economy is to be kept constant the hourly

coal-consumption must be diminished in the same proportion as the grate surface is increased.

b) For a given grate-surface the consumption of coal per hour must vary as the square of the heating surface. Thus, if the heating surface is doubled, four times the coal can be burnt on the grate without influencing the evaporative power.

c) For a given hourly coal-consumption the grate surface must vary as the square of the heating surface if the evaporative power is to remain constant. If the heating surface is doubled the grate surface can be increased four times without altering the evaporative power.

40) From the above proposition a we may infer that for economy the grate surface can never be too small. In practice however the grate cannot be sufficiently reduced to produce the greatest economy because the velocity of evaporation then falls too low. In every boiler the weight of coal which can be economically burnt per sq. m of grate per hour has a certain maximum value and therefore the grate area must be made to correspond with the intended velocity of evaporation, for in general the weight of steam generated per hour is proportional to the weight of fuel burnt in the same time. From propositions b and c we may infer that the evaporative capacity of a boiler increases in a higher ratio than the heating surface if the economy remains constant because by doubling the heating surface the hourly consumption of fuel and water can be quadrupled either by forcing the combustion or enlarging the grate. This inference is however subject to a limit because the weight of fuel which can be economically burnt per sq. m of grate per hour has a certain maximum value which also limits the evaporative capacity obtainable by increasing the grate. But that the above conclusion is in general true is also shewn by DONKIN'S*) experiments. He reduced the heating surface of a tubular boiler 41 % by stopping up some of the tubes and then found that the evaporative capacity of the boiler was only diminished 6 % while the uptake temperature was only 3 ° C. higher than when all the tubes were in operation. From another point of view this experiment is interesting as demonstrating the great importance of direct heating surface, for here only the indirect heating surface was subjected to diminution.

Inferences from these propositions.

*) Engineering. 1892. I. p. 345.

Velocity of draught.

41) **III. The sectional areas for draught** of a boiler depend with natural draught upon the velocity of the gases issuing from the funnel which has been found by direct measurement to be 6 to 7 m per sec. To keep up a steady draught in the fires the air would require to enter the ash-pit at the same velocity if it were not heated up in the boiler to about 300° C. and thus expanded to more than twice its original volume and if the funnel gases did not also contain the products of combustion of the fuel. As a matter of fact the volume of the funnel gases is about 2.3 times the volume of the air entering the ash-pit and therefore the area of the ash-pit mouths could be $\frac{1}{2.3}$ times the sectional area of the funnel if the air were required to travel at the same speed as the gases. In practice however the area of the ash-pit mouths is made about double the sectional area of the funnel as it is more convenient for the firemen to have the speed of the draught moderate and it is thus limited to about 1.5 to 1.7 m per sec. (compare § 20, 32).

Sectional areas.

42) The sectional areas for draught are found to vary considerably in the practice of different engineers as is shewn by a glauce at cols. 48 to 52 of the tables on pp. 597 and 601 in which the areas of the various cross sections are expressed as usual in terms of the grate-area. Below are given the smallest, mean, and greatest values of the principal draught-sections most usually found in Scotch boilers expressed as fractions of the grate-area.

Section	Smallest value	Mean value	Greatest value
Through ash-pit	0.16	0.20	0.25
„ free grate-surface	0.18	0.25	0.37
Over bridge	0.125	0.18	0.21
Through combustion chamber	0.16	0.20	0.25
„ tubes	0.14	0.17	0.19
„ smoke-box	0.13	0.16	0.18
„ uptake	0.12	0.14	0.17
„ funnel	0.10	0.125	0.16

Remarks.

43) It may be remarked that the free grate-surface must always be somewhat greater than the area through ash-pit because the former soon becomes partially choked by clinker &c. The section over bridge in locomotive boilers is on account of the roomy furnaces much greater than in Scotch boilers and generally varies between 0.25 and 0.33, averaging about 0.3. The section through combustion-chamber varies very much according to th eheight of the chamber, for there is a rule that in single-ended boilers the volume of the chamber should about equal the

volume of the furnace above the bars. In double-enders with combustion-chambers common to two opposite furnaces it can be reduced to 0.75 of the combined volumes of the two furnaces above the bars. Great care must be taken in determining the section through tubes. If made too small it chokes the draught as soon as the tubes begin to get dirty; if too large, the sluggish speed of the gases favours the deposit of soot in the tubes thus checking the velocity of evaporation. The area through smoke-box can be rather smaller than the area through tubes as the subdivision of the gases into many separate streams here ceases and the frictional resistance is thereby lessened. The uptake should taper gradually into the funnel so that there may be no sudden transition of the gases into its comparatively narrow section. This should be kept within moderate limits as very large funnels entail considerable weight especially as they are now usually made double. A disadvantage of large funnels is their extensive cooling surface and as they must be painted often their up-keep is costly.

§ 66.

Artificial Draught.

1) **I. Furnace arrangements with induced draught.** It is necessary for keeping up a sufficient natural or chimney draught that the gases should issue from a steamer's funnel, which is low and besides that made of iron, at a temperature about 300 ° C. above that of the air, thus carrying off nearly one-fourth or the whole heat developed from the coals. But a considerable portion of this lost heat can be utilized if an uninterrupted supply of air to the fires is secured by mechanical means. The gases can leave the funnel at a much lower temperature if they are either sucked or forced through the flues. The heat thus gained can be used within certain limits either in the boiler itself by means of a corresponding increase of heating surface or in the uptake for heating either the air on its way to the fires or the feed-water. **Substitutes for natural draught.**

2) The artificial conduction of air to the furnace requires only such a small expenditure of power as to be scarcely worth consideration in comparison with the saving of heat. This is illustrated by the fact that in estimating, only about 3 *IHP* per sq. m of grate are allowed for to produce 30 mm water-pressure in a closed stokehole with a forced-draught fan. **Fan blast.**

The best speed for the fans is 250 to 350 revs. per min. when their efficiency may be estimated at about 40 %. But in calculating the dimensions of the fan-engine the efficiency of the fan should only be taken at 35 %.

Varieties of artificial draught.

3) The artificial air supply may be obtained
by induced draught or
by forced draught.

Induced draught.

4) Induced draught is obtained by sucking the air into the funnel through the grate and tubes as by the exhaust in a locomotive and the steam-jet in some marine boilers. As the steam used in the latter arrangement is lost and must be made up for by supplementary feed, this plan is not adapted for continuous work at sea and is replaced by a fan in the base of the funnel drawing from the uptake and driven by a small engine.

Advantages of induced draught.

5) *The advantages of induced draught*, besides its being more economical to work than the system referred to in 2), are due to the following circumstances, viz.

a) In war-ships it is possible to omit the funnel altogether because the gases drawn up by the fan can be conducted to and discharged at any place in the ship we please.

b) Induced draught fittings are in general simpler and more easily constructed than those for closed stokehole forced draught.

c) Induced draught keeps the stoke-hole cool if the air is drawn from above deck and is more convenient for the firemen than forced draught with closed ash-pits as no special precautions to avoid being burnt by the flame are required when opening the fire-doors.

d) The tubes last longer than with forced draught. This is a very essential point and is corroborated by the fact that the tubes of locomotives which of course work with induced draught have a much longer life than those of similar boilers in torpedo-boats with forced draught. While the flames when *forced* through the tubes, impinge on the tube-plates before splitting up into separate tongues, they divide up some distance off the mouths of the tubes when they are *sucked* from the funnel. The back ends of the tubes are therefore in the latter case less exposed to the risk of burning than when, as in the former case they are directly struck by the points of the flames.

Disadvantages of induced draught.

6) *The disadvantages of induced draught* may be summarized as follows, viz.

a) The fan, placed in the uptake, must be larger than a forced

draught fan on account of the considerably increased volume of the gases.

b) The uptake to take in such a fan must be much larger and heavier.

c) The fan is less efficient when working in hot gases which may easily occur when the boiler is forced, and the wear and tear is greater. This objection can be got over by taking care that the gases do not enter the fan till they are cooled down, that is utilized as far as possible.

7) **II. With induced draught fittings** the necessary cooling down of the gases after they leave the tubes can be effected by using them either for *Induced draught fittings.*

a) heating the feed-water or

b) warming the air on its way to the fires.

8) a. **Martin's induced draught**, fitted as early as 1886*) to the S. S. "Olive Branch" was shortly afterwards adopted on the S. S. "Bléville" in conjunction with a feed-heater by the late Mr. KEMP**), as shewn in Pl. 46, Figs. 1 to 3. The fan in the funnel draws the air through the grate and then the gases through the large feed-heater referred to further on. Above the fan a valve or damper is fitted enabling the boilers to be worked under natural draught. KEMP'S arrangement has not been a success on account of the continued leaks about the heater tubes for which the system of induced draught was not to blame and always worked satisfactorily. MARTIN states that the fan in the funnel of "Olive Branch" though always running in a temperature of 230° C. has not shewn any excessive wear. *Martin's induced draught.*

9) b. **Ellis and Eaves's induced draught*****) (Pl. 46, Figs. 4 & 5) has, instead of KEMP'S feed-heater, an air-heater on HOWDEN'S plan. The air drawn from the upper part of the stoke-hole enters at *A* two systems of tubes, arranged one on each side of the boiler, in which it is heated by the uptake gases passing among the tubes and then reaches the furnace partly through the grate and partly through openings round the fire-door. The gases are drawn from the furnace through the air-heater and discharged into the funnel by the fan which is placed at the side of the latter. The first evaporative trials of this arrangement on land gave such encouraging results that the steamers "Berlin" †) "Southwark", "Kensington", &c. were soon after- *Ellis and Eaves's induced draught.*

*) The Marine Engineer 1887. p. 324.

**) Engineering 1887. I. p. 54.

***) Transactions of the Institution of Naval Architects. London 1894. p. 42.

†) Ibid. 1895. p. 82.

wards fitted with it. The ELLIS and EAVES system is reported to be 15 % more economical in coal than any forced draught system without air-heating at a combustion of 150 kg per sq. m of grate. The North German Lloyd Co. have recently fitted several of their mail-steamers on this induced draught plan.

§ 67.

Forced Draught.

Definition.

1) **I. Pressure of the air.** In the forced draught system of combustion the air enters the furnace at a certain plenum, or excess of pressure above the external atmosphere, generated by a blast.

Recording the pressure.

2) The pressure of the air for forced draught is so small that it is not expressed in atmospheres but in millimetres of a column of water; 10334 mm of water = 1 atmosphere.

Kinds of forced draught.

3) It is usual to distinguish, according to the pressure, between
forced draught proper of a pressure of 30 mm of water and above, and
assisted draught where the pressure varies between 5 and 15 mm only.

Application and object of forced draught.

4) Forced draught proper is applied to the boilers of fast war-ships as cruisers and torpedo-boats in which a high power has to be developed with boilers as small, i. e. as light as possible (compare the table on p. 576). Moderate forced draught is used on merchant ships, either to facilitate a continuous high rate of consumption, to attain economy of fuel, or to enable low-class coals to be utilized.

Forced draught fittings.

5) **II. Forced draught fittings.** The air under pressure may be conducted either into the stoke-hole only, direct into the ash-pit, or partly above and partly below the bars. The two former methods are usually adopted with high air-pressures, the latter for low ones. We may therefore distinguish

a) closed stoke-holes,
b) „ ash-pits,
c) „ „ and furnaces,
d) open „ and closed furnaces.

Arrangement of closed stoke-holes.

6) a. **Closed stoke-holes** are rendered as air-tight as possible. The ventilators and ash-hoist tubes are provided with doors at the bottom. The approaches to the stoke-hole end in air-locks, small chambers with two air-tight doors one of which must be closed before the other is opened. The stokers work under the air-pressure. Although at first they may feel nervous at

being in a completely closed space they soon become accustomed to it, especially when easily opened danger-exits are provided. The men soon learn to appreciate the advantages of a closed stoke-hole which is in fact cool and airy. No inconvenience is experienced from the air-pressure as this is only 50 to 100 mm of water and, as trials of torpedo-boats have shewn, cannot be raised above 150 mm because at this pressure fragments of burning coal as large as hazel-nuts are blown out of the funnels and the tube ends become stopped up at the backs with flying ash and melted clinker forming what are known as "birds-nests".

7) The air from the fan must enter the stoke-hole at as high a position as possible. If the orifice is too low and the current of air strikes the coals on the stoke-hole plates the whole place is filled with coal-dust which is injurious to the eyes and lungs of the stokers. The air enters the ash-pits and penetrates between the bars into the fires. In consequence of the resistance it experiences in passing through the grate and the fuel the pressure falls and is somewhat lower in the furnace than in the stoke-hole so that when the fire-doors are opened the air from outside enters the furnaces. Most modern war-ships and the greater part of the torpedo-boats in all navies have closed stoke-holes. Water-tube boilers which are to be forced can only be worked on this system, the reason of which will be given further on. **Applications of closed stoke-holes.**

8) b. **Closed ash-pits** do not necessitate any deviation from the ordinary stoke-hole arrangements for natural draught. The fan delivers the air through a duct which is connected, air-tight, to the ash-pits. When the air has passed through the fire it still has a certain *plenum*, so that on opening the fire-door flame would issue from the furnace. This circumstance necessitates shutting off the blast before opening the door and at first gave designers a good deal of trouble. Satisfactory and durable fittings for the purpose have now been devised chiefly by **Closed ash-pit arrangement.**

α) SCHICHAU of Elbing for torpedo-boats,

β) WILLANS for steam launches,

γ) FOTHERGILL for merchant steamers.

9) *α. Schichau's closed ash-pit**) (Pl. 47, Figs. 1 to 4) is designed for the locomotive boilers of torpedo-boats. The fire-door is suspended on the shaft *a* which can be rotated by the lever *b*, by means of the hinges *d* and d^1 pressed down by the spring *f*. The hinge d^1 has a catch c^1 which, when the lever *b* is moved far **Schichau's closed ash-pit.**

*) German Patent No. 23581. 1883.

enough, engages with the cam *c* on the shaft and thus opens the door *g*. But before the lever *b* can be turned far enough for the cam *c* to engage the catch c^1, the air-valve *k* must be closed and for this purpose a lever *e* is fitted at one end of the valve-spindle and connected by the rod *i* and the spring *l* with the hand-lever *b*. The ratchet wheel *m* keyed on the shaft *a* keeps the lever *b* in any required position by the pawl *n*. When the blast is to be shut off the lever *b* is placed so as to lift the lever *e* by means of the rod *i* and the spring *l* until the air-valve *k* rests against the two stops h^1 of the air-duct *h*. On moving the lever *b* further, the spring l^1 is compressed in the box *l*, the cam *c* engages the catch c^1 of the hinge d^1 and the door opens, stretching the spring *f*. Immediately the pawl *n* is raised and the hand lever *b* let go, the springs *f* and l^1 shut the door and the valve *k* does not open to admit the blast till afterwards. Both the door and the valve are fitted with asbestos strips so as to close air-tight. The door *p* which can be raised or lowered by the handle *o* closes, when down, the air-duct *h* and enables, by its inclined position, the ashes to be withdrawn from the ash-pit. When getting up steam this arrangement admits the external air to the fire until the fan-engine can be started.

Willans's closed ash-pit.

10) *β. Willans's closed ash-pit**) (Pl. 47, Fig. 5) has a double fire-door, an internal one *F* and an external one *A*. The latter revolves on the shaft *B* and lies when fully open, on the stoke-hole plates. The door *C* is so connected with *A* that it lies up against the top of the air-duct when *A* is shut, but comes into the dotted position and closes the duct as soon as *A* is opened. The internal fire-door proper *F* revolves on the shaft *D* which carries a lever connected by the link *G* to a lever fixed on the shaft *B*. The effect of this arrangement is that *F* cannot be opened until *C* has shut off the blast. The dotted position I shews the furnace open and the blast closed and the dotted position II shews the link *G* when the fire-door is shut. The ashes are very simply got out by drawing forward the lower portion *H* of the bridge-plate. Mr. WILLANS considered it unimportant that the valves and doors should close perfectly air-tight as the plenum is only 10 to 15 mm. The arrangement was fitted to the boilers of fast launches.

Fothergill's closed ash-pit.

11) *γ. Fothergill's closed ash-pit***) (Pl. 47, Figs. 7 & 8) is intended for merchant steamers using poor coal with a good deal of small

*) The Engineer 1885. I. p. 256.

**) The Engineer 1888. I. p. 274.

A sheet-iron casing over the furnace fronts is connected with the fan and so arranged that the portion enclosing each furnace can be removed without disturbing the others. From the casing the air passes to each ash-pit through a separate grid-iron slide-valve. The operation of opening the fire-door catch simultaneously closes the corresponding air-slide by means of a link connecting the slide to a lever keyed on the spindle of the door-catch. So long as the fire-door remains shut the air-slide is open. In other respects the arrangement of the fire-door does not differ from the ordinary one. In order to make the combustion more complete a branch pipe is carried from the air-casing into each combustion-chamber. Each branch is fitted with a cast-iron rose at its orifice to spead the air among the combustion gases and the main duct from the fan as well as the separate branches are all provided with regulating valves. Extra air can be admitted from the ash-pit to the back of the bridge by means of a slide-valve fitted in the lower portion of the bridge-plate. A loose protecting plate is placed in front of this slide valve and provided with eyes for drawing it forward with hooks for the purpose of clearing out the ashes from behind the bridge as in WILLANS'S arrangement. The firebars are 11 mm thick and riveted together in sets of three or four to keep them straight (see Fig. 30 Pl. 42). Behind the bridge a baffle plate *A* is fitted to intercept the gases on their way to the tubes. Wrought iron is found to be the best material for this, asbestos and brickwork having been tried and failed. These baffles will stand about six weeks continuous firing and are then replaced by others made of old ship or boiler plate. FOTHERGILL employs a blast pressure of 75 to 90 mm at the fan in that portion of the blast which goes into the combustion-chamber direct, but the air which gets into the ash-pits expands so much during its passage through the outer casing that it only retains a pressure of 13 to 18 mm. This arrangement has been adopted on some steamers of the Florio-Rubattino Line using inferior North Country coal with a low fan pressure. Although a not inconsiderable increase of economy over the use of superior coal with natural draught has been achieved on this system, it is found that good coal even under a lower blast pressure gives still better results.*)

11) c. **Closed Ash-pits and Furnaces** are almost exclusively applied in conjunction with some device for heating the air on its way to — **Various Arrangements.**

*) Engineering 1887. II. p. 167.

the fires by means of the up-take gases. Of these the best-known systems are

α) HOWDEN'S and

β) WYLLIE'S.

Howden's system.

12) α. *In Howden's System* *) the ordinary smoke-box is replaced by a casing (Figs 1 to 5, Pl. 48) extending according to the diameter of the boiler from about 1 m above the top row of tubes down far enough to cover the ash-pits. The portion which corresponds to the smoke-box itself is fitted air-tight into the casing and nearly touches the outside tubes in each box. The combustion gases pass from the boiler tubes through a nest of vertical tubes, the diameter of which is 89 mm for 76 mm diameter of the boiler tubes, and afterwards through the up-take to the funnel. The air-duct from the fan is connected to the upper part of the casing containing the vertical tubes. The air from the fan passes among and outside these tubes, taking up heat from them, and through the passages between and outside the smoke-boxes where its temperature is further raised, down into the furnaces and ash-pits. Over each furnace door there is a grid-iron valve through which the air passes into the cavity of the hollow furnace front and thence through the perforated baffles into the furnace. The holes in the baffles are horizontal at the lower part and directed downwards at the upper part so that all the jets of air are blown right on to the surface of the fire. Two other valves, one on each side of the ash-pit admit the blast below the bars. By means of the three valves the air supply above and beneath the fire can be independently regulated. HOWDEN prescribes a pressure of about 25 mm above and 10 mm below the bars and assuming the blast orifices to be properly proportioned to the grate, these pressures are stated to produce very complete combustion.

Advantages of Howden's system.

13) HOWDEN claims as the chief advantages of his system the command over the velocity of the blast, allowing of a difference of pressure above and below the fire, besides the heating of the blast. In some of his reported trials the lowest velocity of issue of the blast was about 6 m, the highest about 15 m per second. If the pressure at which the air impinges upon the fire is proportional to the square of its velocity, it follows that the pressure can be varied between 6^2 and 15^2 or about as 1 to 6. By means of the regulating valves a too high pressure above the bars can easily be avoided, so that no obstacle may be placed in the way of the larger volume of air at

*) Transactions of the Inst. of Naval Architects. 1896. p. 182.

lower pressure passing up between the bars. When all is working well the air admitted under the fire serves chiefly to lift the gases liberated by the distillation of the fuel, while the powerful jets of air above the fire penetrate the rising gases in such a manner as to distribute the oxygen necessary for their complete combustion regularly over the grate. Up to a certain point the heating of the blast is found to materially assist combustion.

14) HOWDEN states that by the time the blast has passed through the vertical tubes in the air-heater which are warmed by the up-take gases, its temperature exceeds that of the stokehole by about 100^0 to 110^0 C and is further raised on the passage down the boiler front and round the fire-doors to about 200^0 C on admission to the fires. But as in ordinary sea service it is hardly possible to warm up the blast to such an extent as above, according to trials carried out by SACHSENBERG Bros. in Germany, SPENCE in England, and HOADLEY in America already discussed at length by the author elsewhere*) it appears somewhat doubtful whether the heating of the blast has contributed so materially to the economy of HOWDEN's system as be himself assumes. At any rate the low blast-pressure, the very valuable power of controlling the air-supply, and the high ratio of heating surface to grate are of at least equal if not greater service. HOWDEN's arrangement has chiefly found favour on the larger and faster mail boats, as on the one hand it improves the ventilation of the stoke-hole by withdrawing the hot air from the boiler tops and on the other hand somewhat reduces the comparatively high coal-consumption. The saving as compared with natural draught is estimated at about 15 %.**) HOWDEN's system has been as little employed on ordinary cargo steamers as any other arrangement of artificial draught. It is stated that in such steamers the fan engines, usually much neglected, are very subject to break-downs so that it appears advisable to keep to natural draught. **Value of warming the blast.**

15) *β. Wyllie's System****) has no special air-heater. The hot air from the stokehole and the boiler tops which are surrounded by an air-tight casing is drawn off by the fan and forced through two ducts fitted in the smoke-box into the furnace both above and below the bars. The furnace and ash-pit are each provided with two doors or valves (Figs. 6 to 11, Pl. 48) **Wyllie's system.**

*) Die Entwickelung der Schiffsmaschine in den letzten Jahrzehnten. Edition III. 1892. p. 111 et seq.

**) The Engineer 1893. I. p. 146.

***) Transactions of the Institution of Mechanical Engineers 1886. p. 489.

the hinges of which are geared together by toothed segments. In order to open the furnace or the ash-pit the lever on the spindle of each of the lower valves must be turned towards the boiler, the valve opening inwards then closes the blast orifice from below while the upper valve revolving in the opposite direction covers it above so that it is doubly closed. When the forced draught is working both valves stand vertically and admit the hot air to the fire and the ash-pit. For working under natural draught or to shut off the blast when firing, the valves must be placed horizontally. The blast then comes out into the stoke-hole through an opening, between the fire-door and the damper valve, becoming uncovered by the upper ash-pit valve. The warm blast passes through a sieve-like perforated plate above the fire-door and thus reaches the fire similarly to HOWDEN's. The blast enters the ash-pit through a single opening placed on one side. There is no device for independently controlling the pressure and volume of blast above and below the bars, so that the pressure above is the same as below and not greater as in HOWDEN's system. But WYLLIE admits a portion of the air from the ash-pit through a perforated plate under the bridge into the combustion-chamber to mix with the gases therein. What the pressure and admission temperature of the blast are is not stated. The latter cannot be so high as HOWDEN's as appears by a glance at the devices for heating the blast in the two systems, so that HOWDEN's is certainly the preferable one.

Anderson & Mc. Kinnell's arrangement.

16) d. **Open Ash-pits and Closed Furnaces** as proposed by ANDERSON and MC. KINNELL*) have been fitted on the steamers of several English lines (Figs. 9 to 10, Pl. 47), the whole arrangement being extremely simple. An independently driven air-pump draws air from the holds, stokehole, or engine room, thus ventilating these spaces, and delivers the air through nozzles, usually two to each ash-pit. In other respects the ordinary arrangement of ash-pit is retained. The nozzles are made of bronze and can be adjusted so as completely to regulate the blast. They are enclosed in steel cones *A A*, each of which can be turned round out of the way when the pricker is to be used or the ashes raked out. The blast is closed automatically by the act of thus turning the jet aside so that a loss of blast is avoided. A blast-nozzle *B* is also fitted to the fire-door to assist the combustion. The nozzles are of very simple design and as they have no moving parts, cannot easily get out of

*) The Marine Engineer 1888. p. 177.

order. The powerful current of blast issuing from the nozzles sucks in the surrounding air partly through the steel cones and partly through the open mouths of the ash-pits, producing a very active draught. The arrangement is specially to be recommended for existing boilers which it is desirable to work with moderately forced draught.

17) **III. Advantages and Disadvantages of Closed Stokeholes and Closed Ash-pits.** The closed stokehole has the advantage of not requiring any special precautions when opening the fire-doors. The stokehole is always well ventilated whatever the strength and direction of the wind may be, a particularly important matter in the tropics. All high ventilator cowls which disfigure so many steamers and increase the air-resistance of the ship are unnecessary, as the fans draw the air through trunks carried up to the hurricane deck with mushroom tops. Advantages of closed stokeholes.

18) The greatest objection to the closed stokehole is the rush of considerable quantities of cold air into the furnace when the fire-door is opened for firing and especially for cleaning fires. This current of air, which is very cold in winter in our latitudes and has a sectional area equal to the fire-door aperture, impinges in locomotive boilers directly on the highly heated copper tube-plate and may in a few minutes cool and contract it to such an extent as to cause the tube ends to leak and allow an escape of hot water. Attempts have been made to get over this trouble by carrying the bridge to a considerable height, by fitting a second bridge in the combustion chamber (when there is one), by making the furnaces as long as possible so as to warm up the air somewhat before it enters the tubes, and finally by the introduction of steel tube-plates which are less liable to expansion and contraction than copper ones. In good boilers with well trained stokers these measures have had some success. In return-tube boilers not only the combustion-chambers but the smoke-boxes have been made separate as far up as the base of the funnel. Their mouths are provided with separate dampers conveniently worked from the stokehole. When a fire is to be cleaned its corresponding damper is closed and the passage of cold air through the boiler prevented. At first the fire-bars used frequently to get red-hot and had to be partially renewed after a few hours' steaming under a high plenum, but this difficulty has been completely overcome by making a trough in the bottom of the ash-pit and keeping a few inches of water in it when working under forced draught. Another charge has been made against the closed stokehole Disadvantages of closed stokeholes.

viz. that its ventilation is very defective under natural draught because of the small number of deck openings. This point is not serious in reality, for should the stokehole become intolerably hot it is only necessary to start the fans to fill the stokehole with fresh air after driving the hot air up through these few deck-openings. The plenum thus produced in the stokehole, although hardly worth mentioning, nevertheless causes a difficulty in holding the steam back as is sometimes necessary in war-ships when keeping their place in a squadron at low speed, but in such cases it is easy to take the excess steam to the condensers in order to avoid continually blowing off at the safety-valves. It must however be admitted that getting up the ashes is a somewhat slower process in a closed stokehole than in an open one on account of the opening and shutting of the air-locks at both ends of the ash-hoist tube and this objection is of course more marked when working at high power. It however disappears if ash-ejectors are used.

Advantages closed Ash-pits.

19) Closed ash-pits avoid the trouble from the in-rush of cold air to the furnaces and as while the fire-door is shut the air-pressure is greater inside the furnace than in the stokehole even less external air will pass through the opened door than is the case with natural draught. One special advantage of closed ash-pits is that while cleaning one fire the combustion in the others can be heightened by driving the fan faster and the otherwise unavoidable fall of steam thus prevented. The speed of the ship is further steadied by the comparative ease with which the short fires used with forced draught can be cleaned. With closed ash-pits the air for combustion can be warmed which is impossible with closed stokeholes.

Disadvantages of Closed Ash-pits.

20) It may be objected that in case of accident to the air-valves it is possible to open the fire-door without the blast being shut off and for the firemen to be injured by the issuing flame. Closed ash-pits are also more costly than closed stokeholes on account of the air-ducts, valves &c., besides the ventilating arrangements for the stokehole which are required as much as for natural draught. Finally, closed ash-pits are inadmissible with water-tube boilers, as the combustion gases are forced out into the stokehole through the interstices of the thin casings.

Limit of Blast Pressure.

21) Even in war-ships the blast pressure is no longer kept so high as was the case shortly after the re-introduction of forced draught about ten years ago. Experience has shewn that repeated heavy forcing not only shortens the life of a boiler but also necessitates constant repairs. A plenum of 30 mm

has therefore now become the ordinary limit in war-ships, heightened in extreme cases to 50 mm. Pressures up to 150 mm as formerly applied in torpedo-boats are now completely obsolete. Forced draught always remains a good appliance for increasing the power in case of need. As much as 90 kg. of water per sq. m. of heating surface have been evaporated per hour with a pressure of 125 mm of water in closed ash-pits.

22) With a plenum of 50 mm which is now the ordinary maximum in most navies, a factor of evaporation of 7.5 can be attained in Scotch and locomotive boilers, with an increase of 40 to 50 % over the full power under natural draught. The consumption per *IHP* per hour is at the same time augmented about 10 % by the loss of evaporative efficiency due to the much too high uptake temperature. This means that 45 % more power than the natural draught full power requires 60 % more coal, whence it follows that *with a high plenum the consumption increases in a higher ratio than the power.*

Coal-consumption with Forced Draught.

Twelfth Division.

Marine Boiler Fittings.

§ 68.

Blow-off Arrangements &c.

Stop-valves. 1) **I. Stop-valves** are used to close the orifices of the steam-pipes at the boilers. In marine practice they are usually mitre valves either with cast-iron chests and bronze valves and seats or made entirely of bronze. They are always placed as high as possible on the boiler and arranged so that the steam pressure tends to open them. They have sometimes been designed as double-beat valves, but this is now unusual although such valves are much more easily closed than the ordinary valve but will not keep tight for any length of time. Two stop-valves are generally fitted to every boiler, one for the main steam the other for the auxiliary connections, but both valves are often arranged in a single chest to avoid making two holes in the boiler shell. In recent German war-ships the auxiliary steam-pipes are branched off from the main steam-pipe and connected direct to a few of the boilers only, so that when overhauling the main engine cylinders or the main steam-pipe these can be completely shut off and the auxiliaries run from the few boilers connected to them, their auxiliary stop-valves being entirely separate from the main stop-valves.

Tell-tales. 2) In the German Navy the main stop-valves and all valves in general are provided with *tell-tales* from which it can be seen in the stokehole whether the valves are closed or how far they are open.

Closing-gear. 3) These valves are now also fitted with an arrangement of rods by which they can be opened or closed from the stokehole floor and one of the decks above. Figs. 1 & 3 Pl. 49 illustrate a stop-valve on this plan, combined with a safety-valve.

4) In the absence of a dome or a corresponding contrivance in water-tube boilers, it is usual to fit an *internal steam-pipe* of copper or brass carried along the highest part of the steam-space. This pipe is closed at the end and provided with holes or slits, to diminish priming (Fig. 14, Pl. 49). A small drain-hole is sometimes made at the lowest part of this pipe. **Internal Steam-pipes.**

5) **II. For emptying the Boiler** and for blowing off any mud, dirt, or salt contained in the feed, every boiler is fitted at its lowest point with an aperture which can be closed, usually a cock, called the *drain cock*. In the German Navy these cocks are often fitted as *high pressure cocks* (Figs. 6 & 8, Pl. 49), so that in case of fracture of the gland-studs, the plug cannot be blown out of the shell. **Drain Cocks.**

6) As cocks, particularly when clogged with dirt, are difficult to move and this objection has become more obtrusive with the rise of working pressures, it is now usual to employ *blow-off valves* (Figs. 9 and 10, Pl. 49) instead of cocks. The latest German Admiralty regulations prescribe that only valves are to be used for this purpose. **Blow-off Valves.**

7) In water-tube boilers the blow-off cocks or valves are attached to specially arranged mud-collectors to be more particularly referred to in § 71. **Mud-Collectors.**

8) **III. Surface Blow-off Arrangements.** Lime salts which have become insoluble and muddy particles are not deposited on the boiler bottom or in the mud-collectors until a certain degree of concentration is reached; before this occurs they float in the form of scum on the surface of the water and greatly conduce to priming. For the purpose of removing this scum *scum-dishes* (Fig. 5, Pl. 49) are fitted about 100 mm under the lowest water-level or sometimes inverted just above the surface of the water and connected by pipes to the *scum-cocks* which are attached to the boiler shell in a convenient position. The remarks in 5) also apply to these cocks and the whole arrangement is sometimes called *the surface blow-off* (Fig. 3, Pl. 49). **Scum-dishes Scum-cocks.**

9) In place of the scum-cocks *scum-valves* are used for the reasons and in the cases mentioned in 6). Both valves and cocks are provided with scales to indicate how far they are open. **Scum-valves.**

10) The blow-off and scum-valves of several boilers are generally connected by a range of pipes to a Kingston-valve under the stokehole-floor provided with a *safety-cock* at the end of the pipe. The cocks are worked by a socket-spanner passed through a hole in the stokehole plate and often so arranged that it cannot be withdrawn unless the cock is shut (Figs. 12 & 13, Pl. 49). **Safety-cocks.**

Kingston-valves.

11) **IV. Kingston-valves** are the only ones used on the ship's bottom (Fig. 11, Pl. 49) and are so arranged that the valve can be taken off outwards when the ship is in dry-dock. The valve is prevented from falling out by a loose ring attached inside the mouth by screws. Kingston-valves for suction connections have external gratings to keep out sea-weed and shells. To reduce the injury to the ship's plating by galvanic action due to the bronze of the valve, particularly important in the main blow-off, a zink washer is inserted in the joint on the shell plating and is easily replaced when used up.

Manipulation of the Blow-off.

12) When steam is on the boilers the opening of the valves described in 5) 6) 10) and 11) suffices to blow out the mud deposited on the boiler bottoms together with part of the water. Water-tube boilers, not being much injured by sudden cooling can be entirely emptied in this manner. But this process is to be avoided with Scotch and locomotive boilers as the rapid cooling of the plates and the violent vibration set up during blowing off are apt to start the riveting of the boiler. With these boilers the main blow-off pipes are connected with the bilge-pump suction, or as is usual in the German Navy, with the feed-donkeys by which the water is pumped overboard from the boilers after they have cooled down.

Emptying Water-tube Boilers.

13) When emptying water-tube boilers for cleaning purposes, the water is allowed to run off through the man and mud-holes into the bilge whence it is pumped overboard. In Dürr and Niclausse boilers it is a good plan to slack up the back-end tube covers so that in flowing out the water may carry away the accumulated grease and mud. Branches are however fitted from the blow-off pipes of water-tube boilers leading to the feed-tanks to run off into them the clean fresh water from the boilers when they have been laid off full of water.

Filling the Boilers.

14) The above-described blow-off fittings are used for running up the boilers when the ship is lying in fresh water good enough for the purpose, — provided of course that the boilers are below the water-line.

Air-valves.

15) **V. Air-valves** (Fig. 4, Pl. 49) are fitted at the highest part of a boiler to blow off the air contained in it when getting up steam. These valves are usually fitted at both ends of thwartship boilers to provide against the effect of a heavy list; they are opened when the fires are lighted and not closed till steam issues from them. They are also used when applying the hydraulic test and when filling up boilers that are to be laid off, also to admit air to boilers after blowing out the water so as to prevent the formation of a vacuum inside.

§ 69.

Feed Arrangements.

1) **For feeding the Boilers.** *Main feed-pumps* driven off the engines are used as well as special *Donkey feed-pumps* and more rarely *Injectors* (see Division XIX, § 103, Division XX, § 107 and Division XXXV, § 173). The opinions of engineers differ on the question as to whether main feed-pumps or special feed-pumps worked independently of the main engines are preferable. **Pumps & Injectors.**

2) In the German Navy the system of main feed-pumps has lately been abandoned and the following feed arrangement adopted. The condensed water is taken by the air-pumps from the condensers to the *hot water tanks* where it is heated, cleaned and afterwards passed into the boilers by the special steam pumps placed in the stoke-holes. With water-tube boilers which are all particularly sensitive to impure feed-water feed-cleaners are also inserted in the range of main feed-pipes. In port service donkeys take the water from the condensers or from the feed-tanks and do not deliver it direct into the boilers but into the hot water tanks whence the feed-pumps proper put it into the boilers as when under way. There is besides a *reserve feed-donkey* which is also employed for other purposes than feeding. **Latest system in the German Navy.**

3) In most merchant ships and the older war-vessels there are one or more main feed-pumps, a feed-donkey for use when the engines are stopped, and a hand-pump to replace the donkey in case it breaks down. Injectors are only applied to small boilers for auxiliary or launch purposes, when a hand-pump is fitted to relieve the injector. Figs. 1 and 2, Pl. 50 shew diagrams of the feed-pipe arrangements of a modern cruiser and of a small steamer. The reserve and harbour feed-pumps draw through the pipes *a* and *b* the fresh water, or in the latest practice with water-tube boilers distilled water only, from the feed-water compartment in the double bottom and the supplementary feed is sucked through the pipe *c* from the same compartment into the condensers to make up for the usual losses. **Arrangement in Merchant vessels and the older War-ships.**

4) **Check-valves.** The feed-pipes terminate at the *check-valves* on the boiler shell. There are generally two to each boiler one for the main, the other for the donkey-feed. In the German Navy and in most other vessels they are now double valves, that is the chest contains two non-return valves, one behind the other, and so arranged that when the one next the boiler is screwed **Check-valves.**

down in the ordinary way the other can be overhauled under steam (Pl. 50, Figs. 5 to 7 and 10 to 11).

Internal Feed-pipes.

5) **Internal Feed-pipes** form a continuation of the feed-pipes in the interior of the boiler and conduct the feed-water to a position where it can mingle with a large volume of water without impinging on particularly hot portions of the boiler plating and thus causing leaky seams by sudden changes of temperature. The intention is to heat up the feed-water as much as possible during its passage through the internal feed-pipe and with this object GRAIG'S nozzles are sometimes fitted in the external feed-pipes of tank boilers (Figs. 9 & 10, Pl. 51). They draw hot boiler water into the pipe and mix it with the comparatively cool feed-water. The circulation is at the same time improved (Figs. 1 & 2, Pl. 51). In water-tube boilers the internal feed-pipe conforms as nearly as possible to the direction of the current of circulation peculiar to the boiler.

Hammering in the Internal Feed-pipe.

6) Care must be taken that no steam-pockets can form in the internal feed-pipe as they have the effect of producing a vacuum on the entry of the feed after a stop and setting up violent shocks between the meeting bodies of water on each side of the pocket, sometimes bursting the internal feed-pipe and endangering the feed-pumps.

Object of Feed-heaters.

7) **Feed-heaters.** In describing feed-cleaners in § 71, 10) the necessity is shewn for heating the feed in order effectively to purify it and in 11) a combined feed heating and cleaning appliance is referred to. Sometimes *feed-heaters* are fitted without any view of cleaning but on the one hand to prevent the impact of cool water upon the hot boiler plates and on the other hand to increase the efficiency of the boiler by utilizing heat which would otherwise be lost, to raise the temperature of the feed.

Graig's Nozzle.

8) The first object is achieved by GRAIG'S nozzle fitted in the feed-pipe as described above in 5).

Morison's Feed-heater.

9) A peculiar feed-heating arrangement was introduced by MORISON. It is based on the principle that during the delivery stroke of the feed-pump the pressure in the air-vessel exceeds that in the boiler and continues to impel the feed-water during a portion of the suction stroke, afterwards falling below the boiler pressure. MORISON connects the air-vessel with the boiler by a pipe *D* (Figs. 12 to 14, Pl. 50) having two non-return valves at *B* which allow water to pass from the boiler into the air-vessel while the pressure in it is down and thus to warm up the feed water in the air-vessel.

10) The BELLEVILLE feed-heater or economizer is intended to fulfil the second object above referred to, viz. to raise the evaporative efficiency of the boiler. It consists of a system of tubes similar to the boiler and placed in the uptake. The feed-water passes through these tubes before reaching the steam-drum, see Figs. 14 to 17, Pl. 51, where *a* is the valve of the automatic feed-regulator, to be described further on, and *b* is the float-column. The feed is conducted from the regulating valve into the bottom row of the economizer tubes, passes through these up into a collecting tube common to all the elements of the economizer and thence into the steam-drum. The arrangement amounts to putting part of the heating surface into the uptake and increasing the original heating surface by this amount. As the path of the combustion-gases among the tubes of a BELLEVILLE boiler is short the temperature at which the gases reach the uptake is considerable, which renders the economizer effective. The cold feed is first warmed up by the uptake gases and subsequently exposed to the hotter fire which greatly increases the degree to which the heat is utilized. Belleville Economizer.

11) The efficiency of the BELLEVILLE boiler has been considerably heightened by economizers. British reports give the following consumptions on 30 hour full-speed "natural-draught" trials. Value of Economizers.

Table shewing the effect of BELLEVILLE Economizers.

I. BELLEVILLE boilers *without* Economizers.		II. BELLEVILLE boilers *with* Economizers.	
Name of Ship	Coal per *IHP* in kg	Name of Ship	Coal per *IHP* in kg
Powerful	0.830	Andromeda	0.797
Terrible	0.783	Diadem	0.738
Arrogant	0.961	Europa	0.889
Furious	0.966	Niobe	0.710
Gladiator	0.903	Argonaut	0.733
Vindictive	0.842	Ariadne	0.792
Average	0.881	Average	0.776

The feed-heater thus saves about 12% of the coal-consumption and has therefore become known as the *Economizer* in England, in France *Economiseur*.

12) **Feed-regulators.** In tank boilers the feed is regulated by hand alone through the check-valves described in 4). For water-tube boilers, containing as they do but little water and having a high rate of evaporation, regulation by hand is not sufficient Feed-regulators.

and automatic *feed-regulators* are used, various arrangements of which are shewn in Figs. 3 to 8 and 11 to 13, Pl. 51. They all depend upon the principle of a float which opens or closes the check-valve according to the height of water in the boiler.

Construction of Feed-regulators.

13) These fittings are always provided with a device for increasing or reducing the lift of the valve while the boiler is working, to adapt them to a higher or lower rate of evaporation. Either the fulcrum of the lever which is actuated by the float is shifted and the relative position of the check-valve and its seat thus altered, as in the Germania feed-regulator by the screw *b* Fig. 1, or as in THORNYCROFT'S arrangement the check-valve is directly moved further from or nearer to its seat (Figs. 3, 4, 5 and 13) by the pinion *c a b* and the screw *d* or by the lever *e f g* which has ball and socket joints. In the BELLEVILLE regulator this object is attained by tightening the spring *f* in Fig. 12.

Various arrangements of Regulators.

14) In the BELLEVILLE boiler, as the water level is some distance below the top tubes the float is placed in a separate chamber outside the boiler. THORNYCROFT, SCHULZ, DÜRR and NICLAUSSE generally fit the float and regulating valve in the steam-drum. When considerations of weight and space are comparatively unimportant, it is advisable to place the regulating valve outside the steam-drum for the sake of easy access. In all cases the float must be protected against the violent oscillations of the water.

Regulating Valves.

15) Besides the automatic check-valve, BELLEVILLE fits a valve capable of hand-regulation in the feed-pipe (Figs. 8 and 9, Pl. 50) to relieve the sensitive automatic valve from the shock of the pumps.

§ 70.

Safety Fittings.

Pressure-gauges.

1) **I. Pressure-gauges** are fitted to indicate the boiler pressure and are mostly BOURDON'S spring gauges. They are placed either immediately on the boiler or close to it, or at the starting platform, or in small vessels on deck. They are usually connected to the top of the boiler by a copper pipe of 10 mm bore which can be closed by the *steam-gauge cock* or *valve* (Fig. 12, Pl. 52). Close to the gauge the pipe has a bend, forming a pocket in which the condensed water collects, provided with a drain cock at its lowest point.

2) In the BOURDON gauge the steam enters a spiral or hook-shaped tube of oval section. The effect of the excess of pressure upon its flat sides is to give it a tendency to straighten out and the movement of the end is transmitted by a small link to one arm of a two-armed lever having a toothed sector at the extremity of its other arm gearing into a pinion on the spindle carrying the pointer. Bourdon Gauges.

3) Duplex gauges are now often used and are the rule in the German Navy. They consist of two gauges enclosed in one shell and are marked to 50% above the working pressure which is distinguished by a red stroke. It is usual to place a lamp close to each pressure gauge. The duplex gauges on the boilers have two separate pipes, each provided with a cock, those on the main steam-pipes are however fitted with a single pipe. Duplex Gauges.

4) **II. Standard Pressure-gauges.** Each boiler in German steamers is provided by law with a branch and cock in a suitable position (Fig. 16, Pl. 52) for attaching the *standard gauge* when testing. This generally consists of 2 BOURDON gauges in separate cases adapted to the necessary high pressure. Standard-Pressure-gauges.

5) **III. Water-gauges.** In marine practice water-gauges always consist of gauge-glasses and gauge-cocks or valves. In Germany every marine boiler has at least two water-gauges placed one on each side so that the water-level can be observed when the ship is listed. Water-gauges.

6) It is usual in marine boilers to place the steam-cock of the upper water-gauge pipe as near the crown of the boiler as possible and the water-cock about level with the furnace crowns. The blow off cock of the water-gauge is in connection with the latter cock and is provided with a pipe leading to the bilge. This arrangement of the valves affords a steadier water-level in the glass than if the cocks are placed in the neighbourhood of the water-surface which is of course broken up by the ebullition. It must however here be remarked that when the water-cock of the gauge is thus placed it has been often observed that when steam is up the water shews considerably lower in the glass than it actually is in the boiler. This is explained by the fact that the water in the exposed copper pipe leading to the bottom of the glass is cooler and heavier than the water in the boiler which is mixed with the rising bubbles of steam and thus a smaller head of the outside water is sufficient to balance that inside. It is therefore advisable to place the water-cock on the boiler-shell nearer to the water-level although the water is Gauge-cocks.

then less steady in the glass but it has the advantage of indicating the true height more correctly and guarding against undue encroachment upon the steam-space which is in no case too liberal to begin with. Sometimes internal pipes are connected to the water-cocks and carried to the lower part of the boiler. Placing the lower cock on one side of the glass which was invariably done in the first-described arrangement as a protection against scalding when shutting off for a burst glass is also admissible in the plan now recommended, as a horizontal connection between the cock and the glass does not interfere with its shewing correctly. For similar protective reasons the steam cock is usually (in the German Navy always) fitted to work with a lever and rod from the stokehole floor. Sometimes valves are used instead of cocks, the stems having a very quick thread so that they can be closed with a quarter of a turn.

Details of Water-gauges.

7) The glass is jointed by screwed glands packed with india-rubber rings. The clear length of the glass is about 30 cm in large boilers, its external diameter for low pressures about 28 mm and 22 mm for high pressures. The gauge is usually placed so that the lowest safe water-level is about 100 mm above the bottom nut. In the German Navy the length of the glass is regulated so as to shew 50 mm of water with the lowest level permitted by the rules and the ship listed 8°. The glasses are issued in lengths which comply with this requirement and give 5 to 10 cm to spare. They are fitted with protectors of wire-net embedded in glass as manufactured by the Actiengesellschaft für Glasindustrie formerly FRIED. SIEMENS or sometimes simply by wire-netting; see Figs. 1 to 5 and 8, Pl. 52.

Manufacture of Gauge-glasses.

8) Durable gauge-glasses are very difficult to make as stresses are set up in them during the process of manufacture and they are exposed in use to very abrupt changes of temperature. At pressures of 14 atmos. and upwards or say a temperature of 200° C. the steam decomposes the glass, carrying off the alkaline silicates and leaving silicate of lime. This action is indicated by the fact that gauge-glasses after being some time in use preserve their original thickness at the water end but are reduced to the thickness of paper at the steam end. The best glasses are green in colour, stubborn to fuse, and contain alumina, such as those prescribed for the German Navy, and very carefully annealed. This renders the skin harder i. e. richer in silicic acid and poorer in alkali.*)

*) Zeitschrift des Vereins deutscher Ingenieure 1888, p. 519.

9) The difficulty of manufacturing durable tubular water-gauge glasses led to the invention of KLINGER's water-gauge (Figs. 6 & 7, Pl. 52). The flat glass sides 15 mm thick are not liable to burst like tubes but they are difficult to illuminate. All glasses in time get rough dirty and partially opaque and it is necessary to place the gauge-lamp so as to bring the glass between the light and the eye in order to observe the water-level with certainty. This is only possible with KLINGER'S when the back as well as the front is made of glass. **Klinger's Water-gauge.**

10) OCHWADT'S gauge consisted of a thick glass with a connection to which could be shut off by a cock, the inside of the boiler (Figs. 9 to 11, Pl. 52). This is really the most efficient water-gauge but has not become popular. **Ochwadt's Water-gauge.**

11) **IV. Safety-valves.** To prevent the pressure rising above its proper limit every boiler is fitted with a *safety-valve* which provides an exit for the surplus steam into the atmosphere immediately the assigned pressure is exceeded. **Safety-valves.**

12) A perfect safety-valve would open with the smallest excess of pressure above the normal one and not close again till this is restored. Such perfection is unattainable in practice, as in consequence of frictional and other resistances the safety-valve does not blow till the normal pressure is exceeded and closes before this limit is again reached. This latter defect is due to the fact that the steam issuing from the valve experiences a loss of pressure, thus diminishing the upward load upon the valve which causes it to close and does not allow it to re-open until the pressure has again risen. Valves have therefore been designed so that the surface exposed to the impact of the issuing steam is increased to correspond with the falling pressure, see Fig. 5, Pl. 53. **Arrangements of Safety-valves.**

13) The object of preserving the full pressure under the valve after it has risen from its seat is better attained by the safety-valve shewn in Figs. 6 & 7, Pl. 53 designed by Mr. KLOTZ an Austrian and introduced into practice by Mr. WILSON an Englishman. It will be seen that the valve is hollow and the cavity of it connected to a pipe leading down into the boiler so that the upward load under the valve is independent of the lift. **Klotze's Safety-valve.**

14) The *area of safety-valves* was formerly regulated by law in Prussia but is now left entirely to the designer. In the German Navy the old Prussian rules are still adhered to, but for other marine boilers the British Board of Trade table given on the next page is usually followed. **Area of Safety-valves.**

Table of areas of Safety-valves.

p.	*F*.	*p*.	*F*.	*p*.	*F*.	*p*.	*F*.	*p*.	*F*.	*p*.	*F*.
15	1.250	46	0.614	77	0.407	108	0.304	139	0.243	170	0.202
16	1.209	47	0.604	78	0.403	109	0.302	140	0.241	171	0.201
17	1.171	48	0.595	79	0.398	110	0.300	141	0.240	172	0.200
18	1.136	49	0.585	80	0.394	111	0.297	142	0.238	173	0.199
19	1.102	50	0.576	81	0.390	112	0.295	143	0.237	174	0.198
20	1.071	51	0.568	82	0.386	113	0.292	144	0.235	175	0.197
21	1.041	52	0.559	83	0.382	114	0.290	145	0.234	176	0.196
22	1.013	53	0.551	84	0.378	115	0.288	146	0.232	177	0.195
23	0.986	54	0.543	85	0.375	116	0.286	147	0.231	178	0.194
24	0.961	55	0.535	86	0.371	117	0.284	148	0.230	179	0.193
25	0.937	56	0.528	87	0.367	118	0.281	149	0.228	180	0.192
26	0.914	57	0.520	88	0.364	119	0.279	150	0.227	181	0.191
27	0.892	58	0.513	89	0.360	120	0.277	151	0.225	182	0.190
28	0.872	59	0.506	90	0.357	121	0.275	152	0.224	183	0.189
29	0.852	60	0.500	91	0.353	122	0.273	153	0.223	184	0.188
30	0.833	61	0.493	92	0.350	123	0.271	154	0.221	185	0.187
31	0.815	62	0.487	93	0.347	124	0.269	155	0.220	186	0.186
32	0.797	63	0.480	94	0.344	125	0.267	156	0.219	187	0.185
33	0.781	64	0.474	95	0.340	126	0.265	157	0.218	188	0.184
34	0.765	65	0.468	96	0.337	127	0.264	158	0.216	189	0.183
35	0.750	66	0.462	97	0.334	128	0.262	159	0.215	190	0.182
36	0.735	67	0.457	98	0.331	129	0.260	160	0.214	191	0.181
37	0.721	68	0.451	99	0.328	130	0.258	161	0.213	192	0.181
38	0.707	69	0.446	100	0.326	131	0.256	162	0.211	193	0.180
39	0.694	70	0.441	101	0.323	132	0.255	163	0.210	194	0.179
40	0.681	71	0.436	102	0.320	133	0.253	164	0.209	195	0.178
41	0.669	72	0.431	103	0.317	134	0.251	165	0.208	196	0.177
42	0.657	73	0.426	104	0.315	135	0.250	166	0.207	197	0.176
43	0.646	74	0.421	105	0.312	136	0.248	167	0.206	198	0.176
44	0.635	75	0.416	106	0.309	137	0.246	168	0.204	199	0.175
45	0.625	76	0.412	107	0.307	138	0.245	169	0.203	200	0.174

In this table

p is the Working Pressure in ℔s per sq. inch,

F the minimum area of the safety-valve in sq. inches per sq. ft of grate surface, all in British units.

For forced draught these figures are to be increased in the ratio of the proposed rate of combustion to that under natural draught.

Loading of Safety-valves.

15) Safety-valves are loaded either directly or indirectly with weights or springs. The oldest system of loading is by *indirect weights* (Fig. 8, Pl. 53) and was abandoned on account of the valves lifting with the motion of the ship. Direct loading with weights was next introduced (Fig. 1, Pl. 54) and displaced by spring-loading for the sake of the very considerable saving in weight, amounting

for instance in the case of the protected corvettes of the "Bismarck" Class to about 4 tons. *Direct* spring-loaded valves are now usually adopted (Figs. 1 & 2, Pl. 49 and Fig. 6, Pl. 53). With indirect springs which were formerly much used (Fig. 1, Pl. 53) the joints often got set fast by rust and dirt and the arrangement was abandoned on the introduction of higher pressures. The springs now used are either conical or helical and act either in tension or compression. They are often arranged on the former plan (Fig. 5, Pl. 53) as experience has shewn that tension springs are not so much de-formed in the course of time as compression ones. In all spring-lever safety-valves it must be borne in mind that the leverage of the spring is reduced in the same measure as its tension is increased by the lifting of the valve. To obviate this the fulcrum has sometimes been made capable of shifting. All spring safety-valves are fitted with an arrangement for closing them in case of fracture of the springs. The spring must also be so arranged that it either cannot be further compressed after it is once correctly set or only after taking out a washer. The recent practice in the German Navy is to fit safety-valves with pointers which indicate whether they are closed or lifting and if so how far.

16) The British Board of Trade gives the following formula (in British units) for the size of steel rod for making safety-valve springs; **Calculations for Safety-valves.**

$$d = \sqrt[8]{\frac{WD}{C}} \quad \dots\dots\dots \quad (304)$$

wherein

W = the load on the spring in ℔s.,
D = the diameter of the coil from centre to centre of rod in inches,
d = diameter or side of square of steel in inches,
C = 8000 for round steel,
C = 11000 for square steel.

Springs thus calculated exhibit a compression per turn of coil under the load W expressed with sufficient accuracy for practice by the formula

$$Z = \frac{D^3 W}{d^4 C} \quad \dots\dots\dots \quad (305)$$

wherein Z is the compression in inches

W, D, and d have the same significations as before,
C = 22 for round steel,
C = 30 for square steel.

In the German Navy the following formulæ (in metrical units) are applied

$$d = \sqrt[3]{\frac{W D}{5.6}} \quad \ldots \ldots \quad (306)$$

$$n = \frac{5.7\, W^2 H(-h)}{d^5} \quad \ldots \ldots \quad (307)$$

and

$$D = \sqrt[3]{\frac{1000\,(H-h)\,d^4}{n\,W}} \quad \ldots \ldots \quad (308)$$

wherein

d = diameter of steel in mm,
W = normal load on the spring in kg.,
D = diameter of coil from centre to centre of steel in mm,
n = number of turns of spring,
H = length of spring unloaded in mm,
h = length of spring loaded in mm,

the practice is to make $H-h$ from $0.31\,\delta$ to $0.56\,\delta$, where δ = diameter of the valve.

Tests of springs. 17) Every spring is tested before use. In the first place it is loaded with weights corresponding to the working pressure and the compression is noted. Thereupon the spring is further compressed a distance equal to one-fourth of the diameter of the valve and must shew no permanent set after the removal of the weights. The weight producing the above specified extreme compression must in no case exceed the working load of the spring by 50%. With long springs of large diameter the additional weight to produce this compression is generally under 50% of the working load. The British Admiralty requires that the compression of the spring under the working load shall not be less than the diameter of the valve, so that the increment of load to produce the greatest compression does not exceed 25% of the working load. The spring is always fitted outside the valve chest to keep it from getting rusted by the steam. The Board of Trade passes safety-valves as satisfactory when the accumulation of pressure does not exceed 10% after 20 minutes' sharp firing with everything shut.

Easing-gear. 18) The easing-gear should always lift the valve itself, it is not so good if only the spindle is lifted and worse still to ease up the load only as in this case if the valve happens to be jammed it will not rise. To diminish the risk of jamming, valves guided by pins or tails below the seat are preferable to the winged type, but when these latter are adopted the wings should have about 1.5 mm play for a diameter of 75 mm.

19) The safety-valves of marine boilers are never so absolutely tight at sea, especially in heavy weather, but that a certain quantity of steam passes from time to time up into the waste steam pipe. The condensed water from this steam is usually taken from the safety-valve chest into a tank and used for boiler-feed or washing purposes. **Condensed Water Tank.**

20) When the safety-valves blow off, some condensed and priming water is always carried up the waste steam pipe and causes annoyance on deck. To guard as far as possible against this, *Water-catchers* (Fig. 2, Pl. 53) are sometimes fitted in the waste steam pipe and the intercepted water carried by a pipe to the bilge. **Water-catcher in Waste Steam Pipe.**

21) JUSTICE*) of London introduced some years ago an apparatus for preventing the noise arising from the blowing-off of safety-valves or the exhaust of non-condensing engines (Fig. 18, Pl. 52). It consists of a cylindrical vessel placed longitudinally in the waste-steam-pipe or the exhaust-pipe and contains a throttle-valve which is opened when the apparatus is not in use. At other times it is closed and the steam passes through a copper diaphragm provided with narrow slits, into a space having a similar diaphragm at the other end and filled, in the case of low-pressure boilers, with ordinary glass beads somewhat larger than peas or in the case of high pressure boilers with small hollow brass or copper balls. The steam in passing through this chamber is split up into a number of small streams and effectually silenced. The diameter of the chamber should be from 5 to 6 times that of the steam-pipe and its height about 200 mm. Trials made on the German armoured gun-boat "Biene" and at Portsmouth Dockyard gave satisfactory results. The apparatus is particularly useful in torpedo-vessels with their very high pressures. **Prevention of noise in blowing-off.**

22) In modern practice especial importance is attached to feeding with distilled water and as its production entails expenditure of fuel which must be saved as much as possible, connections have been introduced for carrying any surplus steam direct to the condenser, thus avoiding blowing off at the safety-valves and as these are consequently so rarely in action silent-blow fittings have become almost obsolete. **Prevention of blowing-off of Safety-valves.**

23) Until lately reducing valves were fitted to these surplus steam-pipes leading from the boilers or the main steam-pipe to the condenser to prevent steam of too high a pressure reaching it. But recent experiments have shewn reducing valves to be **Surplus Steam-pipes.**

*) La Marine a l'exposition universelle de 1878. Vol. II. Paris 1879.

unnecessary if the pipes themselves are made smaller than formerly and a wide length of pipe is fitted at the condenser end to reduce the velocity of the steam. The German Admiralty rules require the sectional area of the surplus steam-pipe to be from 1 to 1.3 sq. cm per 100 full power *IHP* and four times this area for the last length of the pipe next the condenser. A baffle plate should of course be fitted in the condenser to protect the tubes from the direct impact of the live steam.

Alarm-valve.

24) **V. The Alarm-valve** as prescribed by the German Admiralty for all tank boilers is a small indirectly weighted safety-valve (Fig. 3 & 4, Pl. 53) fitted at the boiler front so as to be easily observed from the stoke-hole. Its object is to give warning by the hissing of the steam issuing from it when the working pressure is reached in case the steam gauges and main safety-valves are out of order. These small valves are of course very sensitive to dirt &c. and must therefore be carefully looked after.

Automatic Stop-valves.

25) **VI. Automatic Stop-valves.** In § 68, 3) mention is made of an arrangement of rods &c. for closing the main stop-valves from the stokehole floor and from deck. But in case of a fractured steam-pipe this gear is not quick enough to prevent the outrush of such a volume of steam as must be disastrous to all below. To guard against this danger the main stop-valves are sometimes devised so as to close *automatically*. A variety of such plans are extant, based in general upon the principle of attaching the lid of the stop-valve to a piston, one side of which is under the pressure of direct boiler steam and the other side in communication with the steam-pipe which the stop-valve is to close. In case of fracture of the steam-pipe the sudden fall of pressure in it is intended to close the valve. See Fig. 11, Pl. 54.

Requirements to be fulfilled by automatic Stop-valves.

26) Mr. KOCH of Friedrichshafen, the designer of the valve illustrated, thus enumerates the conditions such valves should fulfil*).

1) They must be unaffected by the variations of steam consumption occurring in the regular working of the machinery.
2) They must not close of themselves during warming up or handling the engines ahead and astern.
3) In order not to shut off in case small leaks arise, which although not at first dangerous, appear likely to increase gradually, the stop-valve must be capable of such regulation as to close of itself immediately the leakage exceeds a certain limit.

*) Zeitschrift des Vereins deutscher Ingenieure. 1893, p. 644.

4) The engineers must be able under all possible circumstances to close the valves quickly and conveniently. The apparatus must not refuse during the opening of the stop-valves as experience shews that this is just when accidents to steam-pipes most frequently occur.

5) The valves, having closed automatically, must remain closed long enough for the engineers to secure them by screwing down the spindles in the ordinary way.

6) Means must be provided for quickly re-opening the valves should they by any chance close when not wanted to do so.

7) The automatic mechanism must if possible be so arranged that there are no parts exposed to friction under pressure with consequent wear.

8) The automatic gear must come into action when the valve is opened or closed in the ordinary way by hand so as to guard against its sticking when an emergency occurs.

The illustration shews how these conditions have been fulfilled. The slide-valve *s* establishes connection between the space *A* above the piston *b* and the boiler through the port *m* and with the space *C*, i. e. the steam pipe to be shut off, through the port *n*, and further the space *B* under the piston *b* with the atmosphere. This space *B* can also be opened from deck by means of the small valve *v* which operation shuts the stop-valve quickly. The many conditions with which auxiliary stop-valves must comply show how difficult they are to design and why they have frequently proved failures.

27) The arrangements adopted by HÜBNER & MAYER of Vienna (Figs. 2 to 5, Pl. 54) differs in principle from that described in 25) and introduces a heavy auxiliary valve which is sucked into its seat by the rush of steam occurring on a sudden fall of pressure in the steam-pipe. The valve can also be closed by hand by means of a lever and, which is particularly important, worked up and down on the spindle to prevent its getting set fast with dirt. In this design a stuffing-box, always an objection in automatic valves, is unavoidable. Hübner&Mayer's automatic Stop-valve.

28) In the automatic stop-valve by SCHUMANN & CO. of Leipzig-Plagwitz (Fig. 8, Pl. 54) a stuffing-box for the spindle of the emergency valve is avoided but a piston is introduced which cannot be moved by hand so that there is a risk of its sticking and thus preventing the valve from closing. Schumann & Co's automatic Stop-valve.

29) MAC FARLANE of Glasgow has also introduced an emergency stop-valve in which the valve and its seat are both movable Mac Farlane's automatic Stop-valve.

and the latter is a piston*). As shewn in Fig. 7, Pl. 54, *C* is the stop-valve proper and *B* the emergency valve. On the valve *C* being opened by the spindle the steam presses the emergency valve up and passes through the piston *A* into the steam-pipe. If a fracture in the steam-pipe takes place the opening of the emergency valve *B* is too small to preserve an equal pressure above and below the piston, an excess of load occurs under the piston and pushes it up, thus bringing the seat into contact with the stop-valve. By means of the lever *D* connected with the piston by the sleeve *H* the piston can be pushed back into its original position and also worked up and down to prevent sticking. It is improbable that this device will remain workable for any length of time as the sliding surfaces are almost certain to get clogged with dirt and if the gland is screwed up too tight the sleeve connecting the piston and hand-lever will jam.

Quick-closing valves.

30) **VII. Quick-closing valves.** Automatic stop-valves are not used in the German Navy but so called *quick-closing valves* are combined with the main stop-valves and actuated from various parts of the ship by compressed air. Figs. 12 & 13, Pl. 54 shew a valve of this description as fitted on board "Hela" and "Fürst Bismarck". Here the steam-pipe is not shut off by the main stop-valves themselves but by special valves connected with pistons like automatic stop-valves. These pistons are not worked by boiler steam but by compressed air of high tension carried to all of them by a system of pipes the arrangement of which is shewn by Fig. 2, Pl. 55. *A* is the air-collector, *L* are valves (Figs. 8 & 9, Pl. 55) opened in case of danger by levers. Both ranges of pipes, the red and the blue are under the same pressure. The blue communicates with the back, the red with the front of the large pistons of the quick-closing valves. Upon opening one of the valves *V* of the red range the compressed air escapes from the whole of this range and takes the pressure off the front of the pistons, thus leaving them free to move forward in the direction of the arrow under the unbalanced pressure behind them and close the valves. The pressure in the collector and air-pipes is from 25 to 30 atmos. so that in the apparatus illustrated there is a load of about 500 kg. on the piston when the air is let out of one range of pipes. To assist the efflux of air Schleifer of Berlin devised automatic outlet-valves placed at intervals along the red range which open upon a slight fall of pressure. Their action is illustrated by Fig. 10, Pl. 55.

*) Engineering. 1898. I. p. 395.

31) Another kind of quick-closing valve is fitted on board "Victoria Louise". As shewn in Fig. 9, Pl. 54, the compressed air in this arrangement is only admitted to one side of the piston and not until the valve is to be closed. The range of air-pipes is shewn diagrammatically in Fig. 1, Pl. 55. The air-compressing pump *B* is in one of the forward boiler rooms, a main air-collector *C* is placed in each engine-room and an auxiliary one *E* in each boiler-room. On each side of the ship a range of pipes *D* leads from the main to the auxiliary air-collectors *E* on the same side and to the various stations in the ship from which it is desired to control the quick-closing valves on that side of the ship. *Q*, *Q* are reducing valves allowing a pressure of 5 atmos. in the range of pipes *D* and the auxiliary collectors *E* whereas in the main collectors *C* a pressure of 8 atmos. is kept up. The hand-valves *g* (Fig. 7, Pl. 55) effect the sudden release of the air from the range of pipes *D*. When this occurs the valves *F* close the pipes connecting the main and auxiliary air-collectors *C* and *E* and make connection between the auxiliary collectors *E* and the quick-closing valves *A*, thus causing them to shut. The arrangement of the valve *F* is shewn in Fig. 6, Pl. 54. In the chamber *h* the piston *i* is connected to the slide-valve *k* which overlaps the opening *l* leading to the piston of the emergency stop-valve and in the position as drawn establishes by means of the exhaust cavity *m* and the port *n* a communication between the cylinder *a* of the emergency stop-valve and the atmosphere. Through the pipe *d* the compressed air from the main air-pipes enters the bottom of the chamber, forces the piston *i* upwards and then passes by the piston through the grooves *o* and *p* and through *e* into the auxiliary air-collector *E*. Thereupon the pressure is equalized in the auxiliary collector, the valve *F*, and the main air-pipes and so long as this condition is maintained the emergency stop-valve remains open. But if by the opening of one of the relief-valves *g* the pressure in the main pipes and therefore also under the piston *i* is diminished, this piston is pressed downwards on to the disc *r* thus shutting off the auxiliary collector from the main air pipes as well as the cylinder *a* of the emergency stop-valve *A* from the exhaust port *n*, while simultaneously connection is opened between the auxiliary collector *E* and the cylinder *a* of the emergency stop-valve, so that compressed air enters the cylinder *a* and closes this valve. In order to guard against an accidental closing of the emergency stop-valves in consequence of small leaks in the main range of air-pipes, the slide-valve *k* is provided with a

Quick-closing valves with auxiliary air-collectors.

small valve *s* connected with the piston *i* and capable of a limited movement downwards without affecting the slide-valve *k*. Should small leaks in the main air-pipes cause the piston *i* slowly to descend, the valve *s* opens and air passes through a lateral opening in the slide *k* from the auxiliary collector *E* into the port *t* under which there is a small hole and when the slide *k* has descended far enough to bring this hole opposite the port *l*, the excess of pressure passes slowly off from the auxiliary collector *E* through the exhaust-port *n* into the atmosphere instead of entering the cylinder *a* of the emergency stop-valve. This occasions a slight fall of pressure above the piston *i* and prevents the slide from descending far enough to break the communication between the port *l* and the exhaust-cavity *m*, but on the other hand the piston *i* with the slide *k* is pushed back into its normal or highest position. H_1 is a cock which closes when the air pressure in the main pipe reaches 5 atmos. and is intended to prevent the main air-collector from emptying itself when the air is let out of the main pipe. The small by-pass *J*, of about 1.5 mm bore at the narrowest part, serves to equalize the difference of pressure caused by any slight leaks in the main pipes and the apparatus generally.

Comparison of the two Arrangements.

32) The last described arrangement with auxiliary collectors does away with the double range of pipes required in the other plan referred to in 30) but is considerably complicated by the necessity for the valve *F* and is generally speaking not so trustworthy as the first device although this also requires extremely careful attention. The pistons of the emergency stop-valves must be worked by hand every day to prevent them from sticking and the rods are particularly liable to become clogged with coal-dust.

§ 71.

Cleaning Arrangements.

Various cleaning Arrangements.

1) Among the various cleaning arrangements subsidiary to Marine Boilers we may enumerate

I. Feed-cleaners or purifiers,

II. Arrangements for cleaning the water-side of the heating surfaces,

III. Arrangements for cleaning the fire-side of the heating surfaces.

2) **I. Devices for purifying the feed-water** may be divided into two classes, **Purifying the Feed.**

a) Evaporators or apparatus for producing pure distilled water,
b) Appliances for separating the cylinder-oil from the feed-water combined with the necessary devices for heating it.

3) **a. Feed-distillers.** Whereas formerly in the days of low-pressure only drinking water was distilled on board ship for which purpose NORMANDY'S distiller was almost universally used, it has now, since the adoption of high pressures and more especially of forced draught, become of the highest importance to obtain the purest possible feed-water in order to keep the heated surfaces in the very cleanest condition. The first step in this direction was to limit the use of supplementary feed from the sea to cases of urgent necessity and to carry fresh spring water in some of the compartments of the double bottom as a source of supplementary feed, but even this water leaves incrustations upon the internal surfaces of the boiler which interfere with the transmission of heat to an inadmissible degree so that the system of taking only distilled supplementary feed produced at sea has come more and more into use. This process has besides the great advantage of enabling the stock of fresh water on board to be reduced, thus setting free a certain proportion of displacement for other useful purposes. It is true that for the production of distilled water the consumption of coal is indispensable but as 1 kg of coal will make about 7 kg of distilled water from sea-water, the saving of weight is still considerable. **Distilling Apparatus.**

4) *Arrangement of Feed-distillers.* The various kinds of apparatus now used in distilling sea-water for feed purposes are much more simply arranged than NORMANDY'S distiller for drinking water. They are all based upon the principle of pumping sea-water into a container, *the Evaporator*, and here evaporating it by surface steam-heating. The resulting steam is then condensed in the main condenser or a special auxiliary condenser and either immediately used in the feed or stored in the fresh water compartment, the residual brine left in the evaporator being blown overboard. As the salt precipitated from the sea-water (see § 33, 3) adheres principally to the evaporator coils and considerably impairs their efficiency, they have to be cleaned from time to time. Various arrangements of coils have been tried with the view of saving time and trouble in this operation. **Arrangement of Feed-distillers.**

5) Among the older and most popular appliances are WEIR'S (Figs. 11 & 12, Pl. 55) and YARYAN'S (Figs. 3 & 4, Pl. 55). While WEIR passed the heating steam through his tubes, YARYAN took the water through them and obtained a comparatively **Weir's and Yaryan's Evaporators.**

high evaporative performance, firstly by evaporating the sea-water under a vacuum and pumping the residual brine out of the evaporator with a small pump and secondly by utilizing the heat in this brine for warming up the sea-water to be evaporated. But the efficiency was soon reduced by scale on the heating surfaces and there was great difficulty in cleaning the tubes. WEIR'S evaporator was worked under pressure instead of a vacuum and was provided with scum and blow-off fittings like a boiler but was also very awkward and tedious to clean.

Later Evaporators.

6) Both these devices have now been displaced by others which can be more rapidly and thoroughly cleaned. In these the coils are so arranged as to change their form with every variation of temperature. When after being some time in action the coils become incrusted, the steam is shut off them, the hot brine blown out and quickly replaced by cold sea-water. The contraction of the coils under this sudden cooling loosens the scale. To clean the coils thoroughly they must be taken out (for which purpose various devices exist) and the scale removed by beating and chipping. The condensed water draining from the coils is carried either to the fresh-water compartment or to the feed-heater and the coil drain pipe should be large enough to carry off all the condensed water when the apparatus is working steadily. The evaporator should be kept at a pressure of 0.2 to 0.3 atmos. to avoid priming the sea-water into the condenser and baffle plates are fitted inside the evaporator with the same object. Sometimes a reducing-valve is inserted in the vapour pipe leading to the condenser.

Pape, Henneberg & Co's. Evaporator.

7) In PAPE, HENNEBERG & CO'S. Evaporator (Figs. 3 & 4, Pl. 56) the heating pipes are of oval section, the steam inside tending to make them circular. On shutting off the steam and flooding the coils with cold water they return to their original form and the scale cracks off.

Feed-regulator.

8) The feed of these evaporators is automatically regulated by a float (Figs. 5 & 6, Pl. 55) actuating the feed-cock *H*. The lever *G* serves to work the float and cock by hand to prevent them from sticking. The float cistern is bolted to the evaporator shell and communicates with the inside of it by the top and bottom pipes.

Niemeyer's Evaporator.

9) NIEMEYER'S evaporator (Figs. 6 & 7, Pl. 56) differs from the last in having spiral coils placed vertically side by side but also altering their shape with a change of temperature in order to dislodge the scale. The system of heating pipes consists of the divided pipe *A* and a number of coils *B* so arranged that one end of

each coil is connected to the one half of the divided pipe and the other end of the coil to the other half. The divided pipe works in stuffing-boxes and can be revolved so as to bring all the coils outside the shell of the evaporator when they are to be cleaned, after removing the cover *M*.

10) In SCHMIDT'S evaporator the heating tubes are of spiral form placed horizontally and connected together into a single coil which the steam enters at the top at *h*, the condensed water issuing at the bottom at *i* (Figs. 1, 2 & 5, Pl. 56). The apparatus is provided like that described in 7) with an automatic float *f* to regulate the feed. The pipe *g* is a separate supply for use without the float, *K* is a water-gauge glass, *m* the vapour valve, *n* a safety-valve, *o* a scum-cock, and *p* the brine blow-off cock. The cover is fitted with hinged bolts to facilitate rapid opening and closing for cleaning purposes. **Schmidt's Evaporator.**

11) Evaporators are made of various sizes for outputs ranging from 2 to 60 tons per 24 hours. In the German Navy PAPE HENNEBERG & CO'S evaporators are also used for producing washing and drinking water. The gross production of the evaporators on a German war-ship is calculated by the following formula, **Proper Dimensions of Evaporator.**

$$L = {}^5/_8 \left(\frac{4\,N + 20\,n}{1000}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (309)$$

in which expression

N = the maximum *IHP* of the main engines,

n = the total complement of the ship,

L = the output of all the evaporators on board in tons per 24 hours.

Here $^5/_8$ is a factor of safety because the fittings are tested with Baltic water which has a low degree of saltness, but have afterwards to evaporate the dense water of the Mediterranean and Atlantic besides suffering a daily loss of efficiency until restored by a thorough cleaning. The consumption of supplementary feed is calculated at 4 tons per 1000 *IHP* and the supply of drinking and washing water at 10 litres per head of the crew. Thus a cruiser of 15000 *IHP* with a crew of 600 is fitted with distillers equal to an output of

$$L = {}^5/_8 \left(\frac{4 \times 15000 + 20 \times 600}{1000}\right) = 120 \text{ tons}$$

This output is distributed over two installations of 60 tons each or as these are very large, 3 of 40 tons each. When liquid fuel is used the capacity of the evaporators has to be increased to make up for the loss of water through spraying the oil-burners. A 50 ton evaporator weighs about 3 tons complete.

Feed-cleaners.

12) b. **Appliances for cleaning the feed-water.** In the few remaining cases where salt supplementary feed is used and consequently no particular attention is paid to scale in the boilers, no appliances for cleaning the feed-water are employed, but when the feed is made up with fresh or distilled water it becomes a point of essential importance to free the feed-water of the cylinder oil carried over from the engines. The oil is present in the water in such finely divided particles as to impart a milky appearance to it and to render the immediate mechanical separation of the oil from the water impracticable. This separation can only be effected after heating the greasy water or emulsion. Pure mineral oils are hydrocarbons, chemically neutral, remaining unchanged by the high temperature in the boiler. But such oils are unobtainable in commerce, there being always a proportion of fatty acids present forming fatty acid salts with the metallic oxides in the condenser water resulting from the abrasion of the cylinders and slide-faces. These salts are insoluble in pure water and becoming mechanically united with the particles of oil, are precipitated on the hot surfaces of the boiler in the form of a sticky mass. For this reason the cylinder oil carried over in the feed must be removed from it, or if this cannot be completely accomplished, at least rendered harmless. To prevent the formation of the fatty acid metallic salts soda or lime-water is added to the feed-water in the heater and forms soda or lime salts (soaps) with the fatty acids, making a flaky precipitate instead of a sticky mass. The decomposed residue of the fatty salts is arrested as far as possible before reaching the boiler by filtration through coke, sponge, and cloth.

The Germania Co's. Feed-cleaner.

13) In the German Navy the following system of feed-water purification has become established. In the air-pump delivery a large box or tank is inserted, divided into 3 compartments, the first being a feed-heater, the second a coke filter, and the third a space for the reception of the heated and purified feed-water (Figs. 2 to 4, Pl. 57). From the last compartment the feed-pumps take the feed-water either to the boilers direct or first through another cleaner. (Compare diagram of feed-pipe arrangement Fig. 1, Pl. 50.) Over the feed-heater compartment of the above box is fixed a small tank from which lime-water or a solution of soda is allowed to flow into this heater. The heating is effected by a steam coil having at its end a jet through which the condensed water in the coil passes out into the feed. Part of the greasy matter separated floats to the top and is dipped up from time to time after opening the lid *a*, a further portion of the grease is arrested by fragments of

coke placed between sieves at the bottom of the heater. The purified water rises from the compartment under the filter through the pipe *b* into the space *C*. So much of the water as is not taken by the feed-pumps passes through the overflow pipe *d* down into the fresh-water compartment in the double bottom. The filter can be opened up by the door *e* and the coke is renewed when it has absorbed a certain quantity of oil.

14) In this system cleaners are inserted in the main feed-pipes working on the principle of the feed-water being forced through cloths which arrest the grease. When the cloths have become too dirty they are removed and changed, the feed being taken through a by-pass fitted for this purpose. The dirty cloths are washed with soda. The desirability of keeping the apparatus within the smallest possible dimensions consistent with sufficient filtering surface has led to various devices. In the German Navy the most successful have been RANKINE'S (Figs. 8 & 9, Pl. 57) and PAPE, HENNEBERG & CO'S. of Hamburg (Fig. 5, Pl. 57). **Rankine and Pape, Henneberg & Co's. Feed-cleaners.**

15) The Weser Co. of Bremen has lately introduced a feed-cleaner on RANKINE'S system containing a very large filtering surface in a comparatively small shell (Figs. 6 & 7, Pl. 57). **The Weser Co's. Feed-cleaner.**

16) LUNDKVIST'S feed-heater and cleaner (Fig. 1, Pl. 57) consists of a nest of tubes *B* surrounded by live steam through which the feed-water passes and arranged like an ordinary surface-condenser. The oil is separated at the surface of the heated feed-water in the container *C* and is carried off through the pipe at *g* or removed through the hand-hole *F* while the purified feed-water rises from the bottom of the container *C* behind the division-plate *v* and flows through the pipe *b* into the hot water tank. Branches for a gauge-glass are provided at *i i* and another is fitted below at *p*; *c* is the drain for the condensed water from the live steam and *e* a mud-cock. **Lundkvist's Feed-heater and cleaner.**

17) MAC DOUGALL'S oil-separator (Figs. 10 to 13, Pl. 57)*) exhibits another method of attaining the same object. The idea is to trap the oil by repeatedly changing the direction of the current of the feed water. The oil is intended to collect in the receptacle *c* and is drawn off when the gauge shews a sufficient quantity. The baffle-plates *l* in the bottom of the apparatus are to arrest any mud suspended in the water and enable it to be blown out by the cock *h* or removed through the hand-holes *a*. This apparatus, like the others, will only act efficiently when the feed has been previously well warmed up. **Mac Dougall's oil-separator.**

18) **II. Appliances for cleaning the water side of the heating surfaces.** To afford access to the inside of tank boilers and the steam and **Manholes.**

*) DINGLER'S Polytechnisches Journal. 1895, p. 279.

water-drums of water-tube boilers manholes closed by lids or doors are fitted. In exceptional cases only, the covers of circular manholes are attached by studs to a stiffening ring riveted to the boiler shell (Pl. 58, Figs. 4 & 5). Manholes are usually oval 450 mm $\times$ 350 mm if possible, but are sometimes made rather smaller for want of space. The oval form allows the door to be passed inside the boiler and the steam presses the door against the joint. To compensate for weakening the shell by cutting the manhole, a ring of flat or angle bar is riveted round the hole (Pl. 58, Figs. 15 & 16). In recent practice however. the hole is frequently punched smaller than the finished size and the edges flanged inwards thus strengthening the shell and furnishing on the face of the flange a convenient surface for jointing (Pl. 58, Figs. 1 to 3). The joint is made of gasket soaked in tallow, Tuck's packing, or a plait of hemp and lead-wire. The door is held in place by two dogs and studs. This arrangement is a very safe one as the studs are not exposed to any stress. Every boiler should have sufficient manholes to render each part as accessible as possible. See Plates 19 & 38.

Mud-holes.

19) *Mud-holes* are fitted at places too narrow to be accessible, especially in low and awkward corners where mud from the water as well as dirt and scale when the boiler is cleaned collect. Figs. 10 to 12, Pl. 58 illustrate a mud-hole in a corner of the fire-box of a locomotive boiler for a torpedo boat by SCHICHAU and Figs. 17 & 18, Pl. 58 a mud-hole between the shell and two furnaces of a cylindrical boiler.

Belleville Mud-collectors.

20) In water-tube boilers special mud-collectors are often fitted from which the mud is blown off. BELLEVILLE'S arrangement of these is shewn in Figs. 13 & 14, Pl. 58.

Dürr Mud-collectors.

21) The corresponding fitting for DÜRR boilers is illustrated by Fig. 6, Pl. 58. The feed water descends in the front water-chamber and during its sudden change of direction into the inserted tubes, throws down the precipitable substances into the pocket of the water-chamber, whence they are expelled by the blow-off valve *a*. Any dirt and grease carried through the tubes into the back water-chamber are got hid of through the blow-off valve *b*. The sludge remaining at the ends of the water-tubes can only be washed out when the caps are removed from the back ends of the tubes for cleaning the boiler.

Niclausse Mud-collectors.

22) NICLAUSSE fits mud-collectors both at the feed-inlet (Pl. 58, Fig. 7) and at the bottoms of the water-chambers (Pl. 39, Fig. 11

Cleaning Water-tube Boilers.

23) It is best to clean water-tube boilers immediately after running them out warm while the greasy precipitates on the heating

surfaces are still soft and the oil hot. Part of the grease is then washed out by the water.

24) The mud, oil, and scale left in the tubes after running off the dirty water must then be scraped out, for which the cleaning appliances shewn in Figs. 2 & 13, Pl. 59 are used, the former for narrow curved tubes, the latter for large straight tubes. When, as occasionally occurs, narrow tubes are completely blocked with dirt, this is either washed out with a jet of water under pressure, or if this does not succeed, drilled out with a flexible drill (Pl. 59, Fig. 14). **Apparatus for cleaning Water-tubes.**

25) **III. Appliances for cleaning the fire side of the heating surfaces.** While the fire side of the heating surfaces of box, cylindrical, and locomotive boilers is easily accessible for cleaning by means of brooms, scrapers, and tube-brushes, this is not the case in the narrow spaces between the tubes of water-tube boilers, so that they have to be cleaned from the boiler ends. Cleaning doors are therefore provided through which brushes with long flexible handles or sometimes a steam-lance can be introduced to dislodge the soot from the tubes. Fig. 8, Pl. 32 shews a cleaning door of a THORNYCROFT boiler. It closes with a bayonet joint. In DÜRR and NICLAUSSE boilers the plates supporting the back ends of the tubes are arranged like gratings with an opening over every tube for inserting a brush or steam-lance. These openings are of course closed by suitable lids when the boiler is at work. In NICLAUSSE boilers the steam-lance is inserted into the spaces between the several water-chambers, in DÜRR, HEINE, ORIOLLE, D'ALLEST, and other boilers with large vertical water-chambers some of the screwed stays are made tubular for the purpose. Compressed air can also be used for blasting away the soot. To clean the tubes superficially it is sufficient with closed stokeholes to shut the fire and ash-pit doors and open the cleaning doors in the backs. The steam-lance must only be used while the boiler is at work, otherwise the steam condenses on the cold tubes and forms a sticky substance with the soot. Figs. 8 & 9, Pl. 58 illustrate the above-described grating in the front of a DÜRR boiler and Fig. 1, Pl. 59 the steam-lance. **External cleaning of Boilers.**

§ 72.

Appliances for Special Purposes.

1) **Zink plates** are fitted with bright metallic contact either to the plates, steam-space, or combustion-chamber stays in marine boilers to protect them from injury arising from galvanic currents. In the German Navy plates $300 \times 150 \times 25$ mm are used when **Zink Plates.**

there is sufficient room but in water-tube boilers smaller plates often have to be applied. Their number is regulated so that one plate of the above dimensions is fitted per 10 sq. m of heating surface in boilers with brass tubes and per 20 sq. m in those with iron ones. In water-tube boilers the zink plates are enclosed in baskets (Figs. 24 & 25, Pl. 59) to prevent the oxide of zink produced by their galvanic decomposition from blocking the tubes. The effectiveness of zink-plates will be discussed in the twenty-eighth Division.

Hydrokineters.

2) **Hydrokineters** are employed in locomotive and cylindrical boilers to improve the circulation and warm up the cold water (Figs. 1 & 2, Pl. 51). Steam enters through the valve *a*, carries the water with it on its passage through the nozzle *b*, drives it through the pipe *c* and gives it an impulse in the direction of the arrow on issuing from the orifice *d*. Thus the water is set in motion along the bottom and up the end of the boiler and uniformly warmed up throughout. The hydrokineter is supplied with steam either from its own boiler, thus promoting evaporation, or else it takes steam from another boiler while steam is being got up in its own, thus shortening the process considerably. By using the steam from another boiler steam can be got up in a tank boiler in 1 hour.

Hydrokineter Connections.

3) Besides the stop-valve, a non-return valve is fitted at *a* to prevent water passing from the boiler into the steam-pipes. If these are not connected to the range of auxiliary steam-pipes but to each boiler direct, another non-return valve is fitted close to the stop-valve on each boiler to prevent steam passing from one boiler to another.

Draining arrangement.

4) When the hydrokineter is in use the water in the boiler which is being heated up rises unduly high and to obviate this a drain-pipe is fitted to each boiler leading to the condensed water tanks or fresh-water compartment.

Mark shewing top tubes.

5) **Mark shewing top tubes.** Every German marine boiler has an indelible mark shewing the highest part of the flues in a thwartship direction. In the German Navy a brass strip is screwed on the end plate of locomotive and cylindrical boilers at the level of the combustion chamber crown for this purpose.

Manufacturer's Name-plate.

6) Besides the above every German boiler has a cast brass plate with raised letters attached to the front shewing the maximum working pressure, the manufacturer's name, the works number, the year of completion, and the dimension of height of minimum water-level above the top of the flues.

Fastenings of the Name-plate.

7) This plate is secured in the following manner. Two copper studs are tapped into the boiler end-plate, they are squared

and fit through two square holes in the name-plate, being afterwards riveted over on it and stamped on the points by the government official who passes the boiler.

8) **The steam-whistle** which serves to give signals, especially in darkness or fog, is generally connected to the auxiliary steam-pipe and worked by a lanyard from the captain's bridge. It may utter a whistling or a hooting note, the former being generally used on small, the latter on large vessels. Its principle is that of a "stopped" organ-pipe (Figs. 20 to 23, Pl. 59). The steam issuing from a narrow annular slit sets up vibrations in a thin bell placed above it, and thus produces a note, the pitch of which may be tuned higher by setting the bell downwards and lower by raising it. Organ-whistles (Figs. 17 to 19, Pl. 59) also work satisfactorily. **Steam-whistles.**

9) **Syrens.** Within the last few years syrens and whistles have been used together in the German Navy but recently the former have entirely taken the place of the latter, two being fitted to every ship, attached to different funnels when the vessel has several and worked like whistles from the bridge. Their construction is shewn in Figs. 15 & 16, Pl. 59. The steam passing through the small slots *a*, causes the drum *b* to rotate very rapidly, so that the currents of steam from the slots undergo a series of quickly succeeding interruptions. A shrieking note of very piercing quality is thus produced due to the great number of vibrations imparted to the air. The drum revolves on the principle of a turbine, the guide-blades being the walls of the slots *a*. The apparatus is started by lifting the valve *c* and this movement of its spindle sets the drum *b* in motion by means of the small pall *e*, which is necessary in case the drum should stop in such a position that the slots are blinded and steam cannot pass. In order to limit the revolutions of the drum and keep the note emitted from getting too high, a regulating brake *f* is fitted in the form of a bell-crank lever, one arm of which carries a vane which offers an increasing resistance to the air or steam the faster the drum revolves and thus presses the cam-like extremity of the other arm against the wall of the cylinder *g* and brakes the speed of the drum. The chain-wheels *h* serve to trim the syren into any direction in which it is desired to signal. **Syrens.**

§ 73.

Steam-driers and Superheaters.

1) In the days of low pressures when boilers were fed from the sea wet steam was injurious to the cylinders because the saline **Necessity for Superheaters.**

particles carried over with the priming water were deposited on the slide faces and in the cylinders and gradually distroyed them. Superheaters were therefore originally introduced (in the uptakes) not for the sake of superheating the steam but of evaporating the water carried over. It was not till afterwards when the harmless character of superheated steam was recognized that special superheaters for low-pressure boilers were introduced in which the steam was not superheated more than 30^0 to 40^0 C at the utmost; but these have now become obsolete for the reason given in § 13, 22. Since in recent times large quantities of steam are produced in the smallest possible boiler forced to its utmost capability, the scrupulous cleanliness of the heating surfaces of the boiler has become the chief object of the sea-going engineer's care. Various evaporators are described in § 71. The only sources of salt now remaining are under modern conditions leaky condenser tubes or the priming of the evaporators and its effect even with wet steam is negligible. Cylinder oil is however of greater danger for the boilers. A perfect feed-cleaner has not yet been produced and the more grease the feed-water contains the lower the efficiency of the feed-cleaner falls. Every good engineer endeavours to use as little cylinder oil as possible and it is a not uncommon case to find that engineers who keep their engines in the best condition have done away with impermeators. The use of oil for the cylinders and slides can however only be curtailed when the steam is moist and itself performs the duty of lubricating the surfaces. In spite of the great advantages accruing from the use of dry and superheated steam (see Zeitschrift des Vereins deutscher Ingenieure, 1888 p. 442 and GRASHOF Vol. III p. 542) it is therefore abandoned in marine practice and moist steam has come into use. An excess of moisture only is to be avoided.

Application of Superheaters.

2) High-pressure marine boilers with large steam and water-spaces generate steam of sufficient dryness. Some water-tube boilers are so designed that the steam-space or at least a portion of it, is in the fire, so that superheaters are of no service. This is particularly the case in BELLEVILLE, THORNYCROFT, SCHULZ, HERRESHOFF, WHITE, WARD, MOSHER and ORIOLLE boilers. NICLAUSSE places a dome on the steam drum and has no superheater.

Dürr's Superheater.

3) DÜRR uses special superheaters for the better drying of the steam, constructed similarly to the boilers themselves. As shewn in Figs. 1 to 3, 8 & 9, Pl. 60 there are in both these boilers two chambers and a number of FIELD tubes and they only

differ in this respect that in the one case the chambers are attached to and the superheater tubes connected with the steam drum, and in the other case the superheater is separate from the boiler proper and placed in the uptake. In the second case the combustion chamber between the tubes of the boiler and those of the superheater is roomier than in the first. The combustion-gases can be diverted from the superheater by means of dampers and led past it into the funnel.

4) As mentioned in 2) the prevention of priming is most important and for this purpose *baffles* are fitted in the steam-drums of some water-tube boilers, consisting of a system of bent pieces of thin plate which arrest the water on its passage among the steam towards the stop-valve. **Water-separators.**

5) In DÜRR'S superheater Fig. 8, Pl. 60, the steam, before reaching it, passes at *a* through a series of baffles shewn in Figs. 4 to 7, Pl. 60. The curved guide-plates *b* constrain the steam to impinge on the baffles *c* causing the priming water to fall and flow through the openings in the plate *d* back into the steam-drum. **Dürr's Water-separator.**

6) In THORNYCROFT and SCHULZ boilers it is likewise necessary to get rid of the water in the steam by means of baffles. These are shewn in Fig. 21, Pl. 39 and on a larger scale in Fig. 10, Pl. 60. The several baffles are zigzagged in position. The water thrown down falls back into the boiler. **Thornycroft's Water-separator.**

7) BELLEVILLE has a different arrangement of baffles, as shewn in Fig. 22, Pl. 39. They take up a great portion of the steam-drum as they have to be particularly efficient on account of this boiler's strong tendency to prime. Small holes are necessary in the upper parts of the baffles to allow all air to escape when the boiler is filled right up to be laid off, otherwise the baffles are very rapidly destroyed by rust. **Belleville's Water-separator.**

8) BELLEVILLE employs a further means of drying the steam. The boiler pressure being 18 to 20 atmos., a reducing-valve is inserted in the main steam-pipe bringing down the pressure to about 13 or 15 atmos. at the engines. This sudden throttling of the steam without taking any work out of it effects the evaporation of the remaining watery particles carried over. The valve is illustrated in Figs. 12 to 14, Pl. 60. The piston-valve is kept open by the pressure of the steam under the piston and closed by the pressure of the steam above it aided by the tension of the spiral springs which are adjusted by means of the hand-wheel on the screwed spindle so that the loads above and below the piston balance each other, but if the pressure beyond the valve, i. e. above the piston, rises, the **Reducing valve used as a steam-drier.**

valve is closed. These reducing valves are reported to have worked satisfactorily on the British cruisers "Powerful" and "Terrible" fitted with BELLEVILLE boilers.

Duncan's Steam-drier.

9) DUNCAN of Glasgow designed an apparatus*) for drying steam (Fig. 11, Pl. 60) in which the water shed by the change of direction of the steam at *a* falls into the receptacle *b* and opens the loaded valves *c* when a sufficient quantity has accumulated.

§ 74.

Cleading of Boilers.

Object of Cleading.

1) Marine boilers are covered with various badly conducting materials called *cleading* to protect them from cooling and to diminish as far as possible the radiation of heat into the boiler-rooms.

Various materials for cleading.

2) Among these materials are
   a) *Horse-hair felt*,
   b) *Slag-wool*,
   c) *Leroy's Composition*, a mixture of cow-hair and cement applied to the boiler in the form of a paste which on being warmed sets to a hard covering from 25 to 50 mm thick,
   d) *Kieselguhr* or infusoria earth treated in a similar manner.
   e) *Cork-bricks* made of cork shavings,
   f) *Asbestos* in various forms either as pasteboard, cloth, or mattresses of white or blue asbestos,
   g) *Pouplier's compound*,
   h) *Artificial tufa*,
   i) *Rudenick's impregnated felt*,
   k) *Sheet iron jackets* enclosing a non-conducting air-space round the boiler shell,
   l) *Combinations* of the above.

Ordinary boiler felt.

3) In the German Navy the common loose so-called boiler felt made of horse-hair has shewn itself to be the worst conductor of heat and is always applied to low-pressure boilers. The sheets of felt are stuck on galvanized iron sheets 1.5 mm thick. bent to the form of the boiler and secured with small screws to angle bars pinned on the shell, the felt being next to the boiler and protected by the galvanized iron from wet and external injury.

Comparative trials of boiler-coverings.

4) Common felt however becomes charred under the temperatures of steam at higher pressures and loses its insulating qualities. Experiments made at one of the Imperial German Dockyards

*) Engineering 1888, II, p. 512.

with copper pipes 165 mm diameter and 1950 mm long cleaded with various materials of the same thickness in all cases, viz. 40 mm, and kept under the same pressure of steam for 2 hours, gave the quantities of condensed water stated in col. 5 of the following table, the quantity for ordinary boiler felt being taken as unity. From these figures the efficiency ratio for each material *at equal thickness* is obtained and recorded in Col. 6. The weights in kg per sq. m are given in Col. 3 and in Col. 4 the cost in marks (say shillings) at date of trial and at the particular dock-yard as an *approximate* guide. The cost of applying the covering to the boiler is however included only for the sheet iron coverings particularized in lines 7, 8, and 9.

Table relating to various cleading materials.

	Character of Material 40 mm thick	Weight in kg per sq. m	Cost per sq. m in Marks	Condensed water	Efficiency ratio
1	2	3	4	5	6
1	Ordinary Boiler-felt without protection .	7.112	3.90	1.000	100.0
2	Cork brick	16.120	9.95	1.174	85.0
3	RUDENICK's impregnated felt	12.500	11.00	1.276	78.0
4	Blue asbestos mattresses stuffed with blue asbestos fibre	6.250	24.13	1.320	75.7
5	White asbestos mattresses stuffed with infusoria thread	22.270	20.00	1.338	74.7
6	POUPLIER's composition	17.720	1.65	1.407	71.0
7	Two thicknesses of sheet iron with dry POUPLIER's composition	45.000	42.12	1.436	69.6
8	The same without POUPLIER's composition	28.000	40.00	1.459	68.5
9	The same filled with infusoria	43.000	42.50	1.471	68.0
10	White asbestos mattresses stuffed with asbestos	19.000	20.17	1.518	65.9

5) Professor CARPENTER read a paper at a meeting of the American Society of Heating and Ventilating of New York on some trials conducted at Cornell University with variously protected steam-pipes which shewed the following losses of heat, that of a naked pipe being taken at 100.

Other Experiments of the sort.

Pipe painted light grey 126.7
Pipe coated with asphalt paint 113.5
Naked pipe 100.0
Two layers of asbestos paper 77.7
One layer of asbestos sheets 59.4
Four layers of asbestos sheets 50.3
Magnesia applied in paste 22.4
Slag-wool felt 20.9
Asbestos and wool felt 20.8

Fibrous slag-wool	20.3
Asbestos and sponge	18.8
Double octagonal wood casing	18.0
Two layers asbestos paper 2.5 mm hair felt between	17.0
The same covered with canvas	15.2

Cork-brick.

6) According to the table in 4) cork-brick is the worst heat conductor of all the cleading materials and at the same time light and cheap. But the bricks are attached to the boiler with a layer of adhesive material about 10 mm thick and are very liable to break on being removed for the purpose of examining the shell externally, thus causing much dirt.

Pouplier's Composition.

7) These objections are still stronger in the case of POUPLIER'S composition as this has to be entirely destroyed when the shell is to be surveyed externally. In other respects this composition may be counted among the worst heat-conductors and the cheapest of boiler coverings.

Leroy's and other coverings.

8) Similar conditions prevail with LEROY'S and other compositions applied wet, among which a mixture of infusoria earth and asbestos fibre has been successful.

Rudenick's impregnated felt.

9) RUDENICK'S impregnated felt is a bad heat-conductor and comparatively cheap and light, but is also injured by removal as the sheets become stiff and break when bent.

Asbestos mattresses.

10) Asbestos mattresses stuffed with asbestos fibre or infusoria thread are not quite such bad conductors as cork bricks and RUDENICK'S felt but they can be easily removed for survey without being destroyed. Blue asbestos mattresses stuffed with blue asbestos fibre are also the lightest of all cleading materials and for this reason are exclusively used in the German Navy for high-pressure tank-boilers. Their high cost will probably hinder their introduction into the mercantile marine.

Usual Cleading in Merchant Ships.

11) The materials most used in merchant ships are LEROY'S and POUPLIER'S compositions and the various mixtures of infusoria earth, asbestos fibre, cow-hair, horse-hair, &c particularly when the circumstances admit of the avoidance of too frequent external surveys.

German Admiralty Rules.

12) The German Admiralty Regulations for boiler cleading, torpedo-boats excepted, are as follows:

a) *Boilers of less than 4 atmos. W.P.* are to be covered with ordinary boiler felt attached by clips to galvanized iron sheets from 1 to 1.5 mm thick.

b) *Boilers of over 4 atmos. W.P.* are if practicable, to be covered with blue asbestos mattresses 40 to 50 mm thick laid over narrow strips of galvanized sheet iron at certain distances apart and punched with holes so as to preserve a thin

air-space between the boiler and the cleading. The bottoms of cylindrical boilers which are liable to contact with bilge-water are not to be cleaded. The mattresses are about 1 m wide and in lengths so adapted to the circumference of the boiler that the joints fit closely. At the boiler ends and the finish of the bottom, angle bars are to be fitted. The mattresses are to be secured by straps alone. The end joints of the various mattresses are to be covered by broad straps with longitudinal perforated sheet iron strips beneath them. In way of man-holes, mountings, &c. corresponding holes are to be cut in the mattresses and carefully stitched round. To protect them from wet and blows and in the neighbourhood of possible leaks from decks and pipe-ranges, a covering of thin galvanized sheet iron is to be fitted. Surfaces which cannot be covered by mattresses are to be cleaded with POUPLIER'S composition with three coats of coal tar applied after it is dry. The mattresses are not to be painted anywhere.

c) *Launch boilers* are to be cleaded either with blue asbestos mattresses, or cork bricks attached by a layer of POUPLIER'S composition 10 mm thick, or with RUDENICK'S impregnated felt applied to a similar coat of POUPLIER. In any case the cleading is to be completely covered with galvanized sheet iron 1 to 1.5 mm thick.

d) Boilers are not to be cleaded till after their first steam trial.

e) The hollow casings of *water-tube boilers* in the stoke-hole are to be filled with infusoria earth, slag-wool, POUPLIER'S composition, or a mixture of infusoria earth with asbestos fibre or pure asbestos wool according to the preference of the boiler manufacturers.

Division XIII.

The Fitting of Boilers on board.

§ 75.

Position of the Boilers in the Ship.

Placing the Boilers on board.

1) **I. Boilers are placed either**

I. *Fore and aft* i. e. with their axes parallel to the middle line of the ship, or

II. *Athwartships*, i. e. with their axes square to the middle line.

Fore and aft Position.

2) I. *In the fore and aft arrangement* the boilers stand side by side between the side bunkers and are fired

a) in small ships, from the engine room, or

b) in large ships from one or more stokeholes separated by watertight bulkheads or cross bunkers from the engine-room.

Athwartship Position.

3) II. *In the athwartship arrangement* the boilers stand

a) side by side in two fore and aft rows with the stokehole between their fronts, or

b) with their backs close together in the middle line and the stokeholes in the wings. This arrangement is frequently adopted in ships with middle-line bulkheads.

Combined arrangement.

4) In large ships a combination of these arrangements sometimes occurs.

Selection of a Boiler arrangement.

5) The decisive factor in settling an arrangement of boilers is the utilization of the available space and the considerations of position and accessibility of the bunkers, sometimes also the design of the boilers, as most of the straight-tubed water tube boilers can only be placed fore-and-aft because the slope af the tubes would cause some of them to get red hot if the ship had a heavy list. Boilers are generally placed amidships forward of the engines although some of the boilers are occasionally placed abaft the engines when considerations of space and trim necessitate it.

Accessibility of the Boilers.

6) If possible the boilers should be so arranged that they have a clear space 300 to 400 mm deep all round them for accessibility

for repairs. In case of necessity they are placed closer to each other and to the ship's sides, but a distance of 100 to 150 mm between the boilers and the bunkers must be preserved. If necessary portions of the bunker plating are made removable. Heavy objects are never to be placed on top of the boilers under the deck.

7) In most ships the auxiliary boilers are placed in the stokehole, sometimes rather higher up than the main boilers. In armoured vessels and large transatlantic steamers they are occasionally put in the tween-decks and in some merchant steamers on the upper deck. In recent times the port consumption of steam for auxiliary machinery, especially electric light plant, has on large ships become so considerable as to require one or two main boilers to cope with it so that in many cases auxiliary boilers are altogether dispensed with. **Position of Auxiliary Boilers.**

8) **II. Boiler Seating.** In way of the boiler-rooms the hull of the ship is specially adapted for the reception of the ponderous boilers by closer spacing of frames, increasing the thickness of floors and stringer plates, stiffening the double bottom, and sometimes by fitting extra side keelsons. Upon this structure the boiler seats are then built to suit the form of the boilers. **Boiler Seating.**

9) *For the old flat-bottomed box-boilers* in wooden ships a flat bed of heavy baulks of timber was provided. The boilers were raised on wedges about 50 to 75 mm above the bed and the intervening space filled with putty which on hardening effectually protected the boiler bottoms from the bilge-water (Figs. 1 & 2, Pl. 19). **Seating for Box Boilers in wooden Ships.**

10) *Box boilers in iron ships* were placed on girders of open or of box form constructed of plates and angles and unless the bilge was particularly deep the boilers were made on the dry-bottomed plan (Figs. 3 & 4, Pl. 19) because with shallow bilges the bottoms of flat-bottomed boilers were inaccessible for cleaning &c. **Seating of Box Boilers in iron Ships.**

11) *For cylindrical boilers and water-tube boilers with cylindrical bottom drums* (THORNYCROFT, SCHULZ, NORMAND, YARROW &c.) the seats are also built of plates and angles and cradle-shaped so as to embrace a portion of the circumference of the boiler shell (Fig. 3, Pl. 20; Figs. 3 & 8, Pl. 22; and Fig. 2, Pl. 61). The boiler is however not bedded directly on the seat but cast or wrought iron chocks are inserted at intervals so that the shell can be cleaned and painted between them. Sometimes feet made of plate are riveted on the boiler shell and arranged to rest on flat-topped bearers (Figs. 1 & 5, Pl. 62). In this case also fitting pieces are inserted between the feet and the bearers. **Seating of cylindrical Boilers and Water-tube Boilers with cylindrical Bottom Drums.**

Seating of Locomotive Boilers.

12) The barrels of these are supported in seats resembling those of cylindrical boilers and the fire-boxes on flat bearers (Figs. 3 & 4, Pl. 61).

Seating of Water-tube Boilers with box-shaped Water-chambers.

13) These boilers, which include DÜRR, NICLAUSSE, BELLEVILLE, HEINE, ORIOLLE, D'ALLEST, and MOSHER boilers, are placed with the water-chambers or the frame-work round the tubes on flat-topped bearers of suitable shape (Figs. 1 & 2, Pl. 31, Figs. 8 & 9, Pl. 33, Figs. 6 & 9, Pl. 62).

§ 76.

The Securing of Boilers in their Position.

The Securing of Boilers on the Seats.

1) The system of *securing the boilers* on their seats must be devised with due regard to the expansion of the boilers by heat. While the distances between the various bearers remain approximately unaltered, the dimensions of the boilers and consequently the distances between the bearing places are increased by some millimetres. The holding-down bolts (Figs. 3 & 5, Pl. 62) are therefore only tightened up sufficiently to prevent the boiler from lifting and to permit of the foot sliding on the bearer, the holes in the foot being made oval with this object. Any greater movement of the boiler than that due to expansion is prevented by chocks or toe-plates, i. e. brackets formed of plates and angles (Fig. 1 f and g, Figs. 3 & 4 c, Pl. 61 also Figs. 6 and 8 c and d, Pl. 62), or by angle-irons (Figs. 3 and 5, a and b, Pl. 62) riveted or bolted to the bearers or the bulkheads. Instead of being bolted to the bearers boilers are sometimes secured by stays attached by forked joints to the bearers and boilers, thus permitting a certain amount of movement (Figs. 1 and 2 a, Pl. 61). The barrels of locomotive boilers are secured to the bearers by straps (Figs. 3 and 4 d, Pl. 61).

Staying of Boilers in the Ship.

2) In spite of the above described fastenings it would still be possible that the great inertia of the boilers might shift them in a heavy sea or when ramming. It is therefore usual to stay them to each other at the tops and also to stay them to or shore them from the beams or bulkheads (Figs. 1 & 2 e, Figs. 3 & 4 a & b, Pl. 61; Figs. 6, 7, & 10 to 15, a & b, Pl. 62). The staying must be effected both in the fore-and-aft and athwartship directions and due allowance must be made here as in the other fastenings for the expansion of the boilers when warmed up. For this purpose the pins in the fork joints of the stays are left a little slack.

3) Brackets or toe-plates (Fig. 4, Pl. 61) must also be fitted so as to leave a certain amount of play equal to the estimated expansion of the boiler and are only to be used in places where the distance from the boiler to the iron-work of the hull is small, for greater distances tension stays are preferable. **Best Arrangement of Stays.**

4) When boilers are stayed to or shored from bulkheads these must be suitably stiffened to prevent their bulging. The bulkheads are sometimes stayed to the bearers as exemplified by the stay *e* in Figs. 1 & 2, Pl. 61 taken from "Wörth". **Stiffening of Bulkheads.**

§ 77.

Stokehole Arrangements.

1) The special arrangements in the stoke-hole for carrying on the work may be classified as relating to **Stokehole Fittings.**

I. Communication,
II. Ventilation,
III. Coal transport,
IV. Ash transport,
V. Stowage of firing tools.

2) **I. The means of communication** are ladders, steps, platforms, and doors. Above the stoke-hole bilges a light frame-work of angle-iron is erected on which the *stoke-hole plates* of checquered plate are laid. These are made thicker in front of the fires than elsewhere, because the large pieces of coal are broken up there. In war-ships the plates at those parts are 10 mm thick whereas 5 to 6 mm is enough for the rest. The stokehole floor should be placed at such a height that the fires can be conveniently worked, 700 mm from floor to bottom edge of fire-door is a good dimension. If the fire-doors are at various heights, as in cylindrical boilers with three furnaces, the height of the lowest should if possible not be less than 420 mm and that of the highest should not exceed 1100 mm. With cylindrical boilers having four furnaces the only plan is to place the stokehole floor lower for the two centre furnaces than for the wing ones, awkward as this arrangement is. If there are passages between or alongside of the boilers, they are laid with plates like the stokehole. **Communication.**

3) When the boilers are large a platform consisting of the usual flat bar frame with round rods is often fitted along the middle of the stokehole above the smoke-box doors for access to the water-gauges, stop-valves, safety valves, and fans. Gratings or planks are often carried along between the boilers at a con- **Platforms, Gratings, Planks.**

siderable height above the floor for getting at the fittings on the tops and to furnish a means of communication between the spaces at the back and front of the boilers.

Ladders and Stairways.

4) Fixed or movable ladders lead from the floor to these landings. In small ships a ladder or stairway is fitted from the stokehole direct to the deck, in large ships to the tween-deck. When forced draught on the closed stokehole principle is adopted these ladders are enclosed in air-tight shafts forming air-locks and fitted with water-tight doors at top and bottom.

Doors.

5) Watertight doors secured with wedge-catches are likewise provided when the stokeholes are separated from each other and from the engine room by water-tight bulkheads, although these spaces communicate with each other. Such water-tight divisions are unusual in wooden ships. Besides the above mentioned doors, sliding water-tight doors leading from the stokeholes to the bunkers are provided in war-ships and merchant vessels of any considerable tonnage.

Ventilating arrangements.

6) **II. Ventilating arrangements.** The fresh air necessary for the combustion of the fuel and the cooling and ventilation of the stokeholes is conveyed thither either by ordinary *cowl ventilators* (Figs. 1 & 2, Pl. 64) carried down as nearly as possible in a vertical direction into the stokeholes, or fans are fitted in addition which draw the air down the ventilator shafts and drive it into the stokehole and the fires. The cowls are fitted so that they can be turned round or trimmed to the wind. When fans are used the shafts terminate in *mushroom heads* instead of cowls, but as experiments have shewn that these mushrooms when fitted to screw down, offer more resistance to the entering air than cowls, they are now often made so as to open like lids. Sometimes doors and openings fitted with sliding shutters are cut in the ventilator shafts at the upper deck to assist the suction of the fans (Fig. 4, Pl. 63).

Rotatory Fans.

7) In the German Navy *rotatory fans* are so proportioned as to deliver 20 cbm of air per kg of coal burnt at the maximum power of the engines, one kg of coal being allowed per *IHP*. The velocity of the air is not to exceed 6 m per sec. on the suction side of the fan and 10 m in the fan case and delivery ducts.

Doors and Valves for Ventilating Shafts.

8) The suction ducts of stokehole fans can be shut off to render them still available when the fans are stopped. When two fans serve one boiler-room and one of them has to be stopped, the compressed air can escape through the other. To prevent this, throttle valves are fitted in the suction shafts or the orifices of the fans are closed by placing a canvas cover over them.

Where these shafts pass through armoured decks, armoured gratings are fitted in them. The designing of fans and their engines will be treated of in § 179.

9) **III. Coal tram-ways** are only found on the larger war-ships. When there are two or more stokeholes often containing upwards of 30 fires it is difficult without some special appliances to keep up the necessary supply of coals from the bunkers. It is therefore not unusual in war-ships to fit rails made of flat bars rather over the height of a man above the stokehole floor and carried from a distance inside the bunkers along the boiler fronts with travellers to which the coal buckets are hooked. Where these rails pass watertight bulkheads they are interrupted by taking out short pieces when the watertight doors are to be closed. Merchant ships sometimes have trucks running on ordinary light rails for the same purpose. **Coal transporting Arrangements.**

10) **IV. All steamers are provided with ash-hoist** gear for getting the ashes and unburnt residne of fuel overboard every 4 hours. This is effected either by heaving the ashes up to the upper deck through a wrought-iron *ash-hoist* or throwing them overboard by an *ejector* (Fig. 5, Pl. 63), through a cast-iron pipe carried from the stokehole through the ship's side above the water-line. In either case the ashes are cooled with water from a special connection for the purpose. They are then put into the ash-buckets and hove up or shovelled into the hopper of the ash-ejector. In the ash-hoist at the upper deck a small winch is fitted, worked either by hand or by a small engine, the *steam ash-winch.* The ash buckets are carried to the side or transported thither on small railways similar to those for the bunkers and tipped out through the ash-shoots fitted in the top-sides. The ash-hoist shafts are provided with a safety arrangement at the bottom to prevent accidents from falling buckets. **Ash-hoist gear.**

11) The construction of the ash-ejector is shewn in Figs. 1 to 3, Pl. 63. A steam pump, one of the reserve donkeys in German war-ships, keeps up a very powerful jet of water from the nozzle of the ejector, sucking the ashes out of the hopper and carrying them overboard up the discharge pipe. To prevent the water standing in the pipe from flowing into the stokehole when the pump is stopped Mess[rs.] HOWALDT fit a jointed lid to the hopper and the Germania Co. a cock with a non-return valve at the bottom of the hopper. The cast-iron discharge pipe is very rapidly destroyed at the bend by the violent friction of the ashes, so that it is necessary to fit loose segments of adamant at this part. **Ash-ejectors.**

Auxiliary Ash-hoist.

12) Ash-ejectors can of course only be used when steam is up and as this is not the case in all the boiler-rooms of a large ship in port, auxiliary ash-hoists must be fitted.

Fighting. Ash boxes.

13) Before ash-ejectors were applied large battle-ships had *fighting ash-boxes* built into their bunkers to receive the ashes during action. They had double walls about 100 mm apart with water circulating between them. The water entered at the bottom from a Kingston-valve and drained off to the bilge at the top thus protecting the bunkers from the heat of the ashes. These boxes contained as much as 40 tons in the earlier German iron-clads of the "Sachsen" Class.

Stowing Firing-tools.

14) **V. The fittings for stowing the firing-tools** consist of semicircular hooks of flat iron attached to the stanchions in the stokehole about 2 m above the floor in handy positions.

Index of Names.

46*

General Index.

C.

Index to Tables.